POWER

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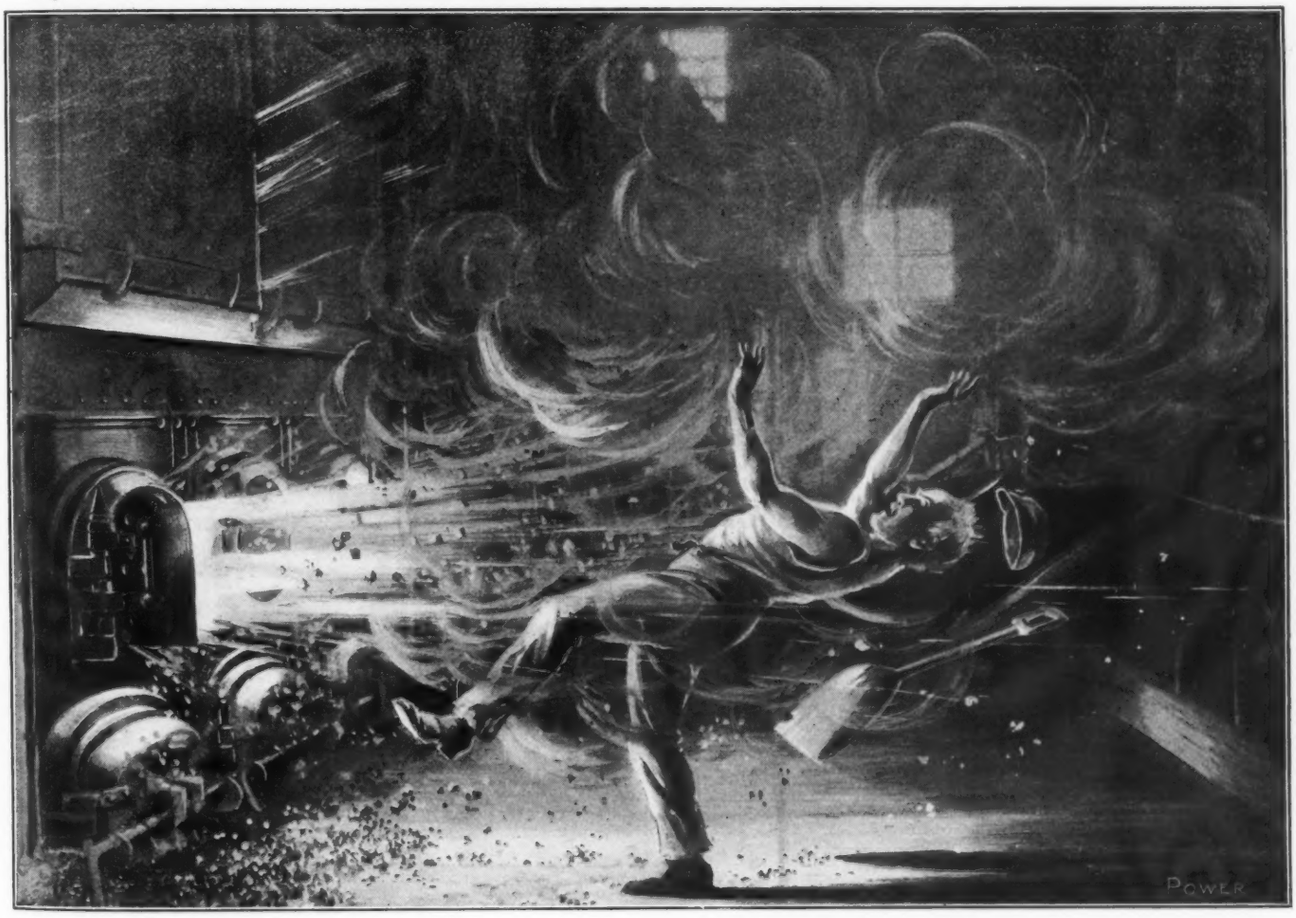
Who Is Responsible?



REPORTS are received every few days of boiler tubes bursting, blowing open the furnace doors and severely if not fatally injuring one or more firemen. Why endanger life from this cause when a simple safety appliance could be so easily attached to the furnace front?

Germany has enforced a law prohibiting placing boilers fitted with steam drains beneath rooms occupied by workmen. There is also a law demanding that all boiler furnaces must be equipped with some type of door that will resist the force exerted against them when a tube bursts.

Good for Germany! But what is America going to do about this avoidable danger?

Operation of Wisconsin's Capitol Plant

Equipment of plant, operating records and costs, and apportionment of initial investment and operating and fixed costs between the various services of the plant.

In these days of active competition between the central station and the isolated plant, it is essential to keep an accurate system of records so as to know exactly the cost of power, and to compare present with past performance. An annual report insures that this comparison will be made and is as instructive to the engineer as it is valuable to the owner. Perhaps at no other plant in the country are the records more complete in every detail than at the plant furnishing heat, light, power and water for Wisconsin's new capitol building and grounds. Every year a complete report is made by John C. White, chief operating engineer, to the superintendent of public property, and, for 1911, that part referring to operation and apportionment of operating and fixed costs may be of general interest.

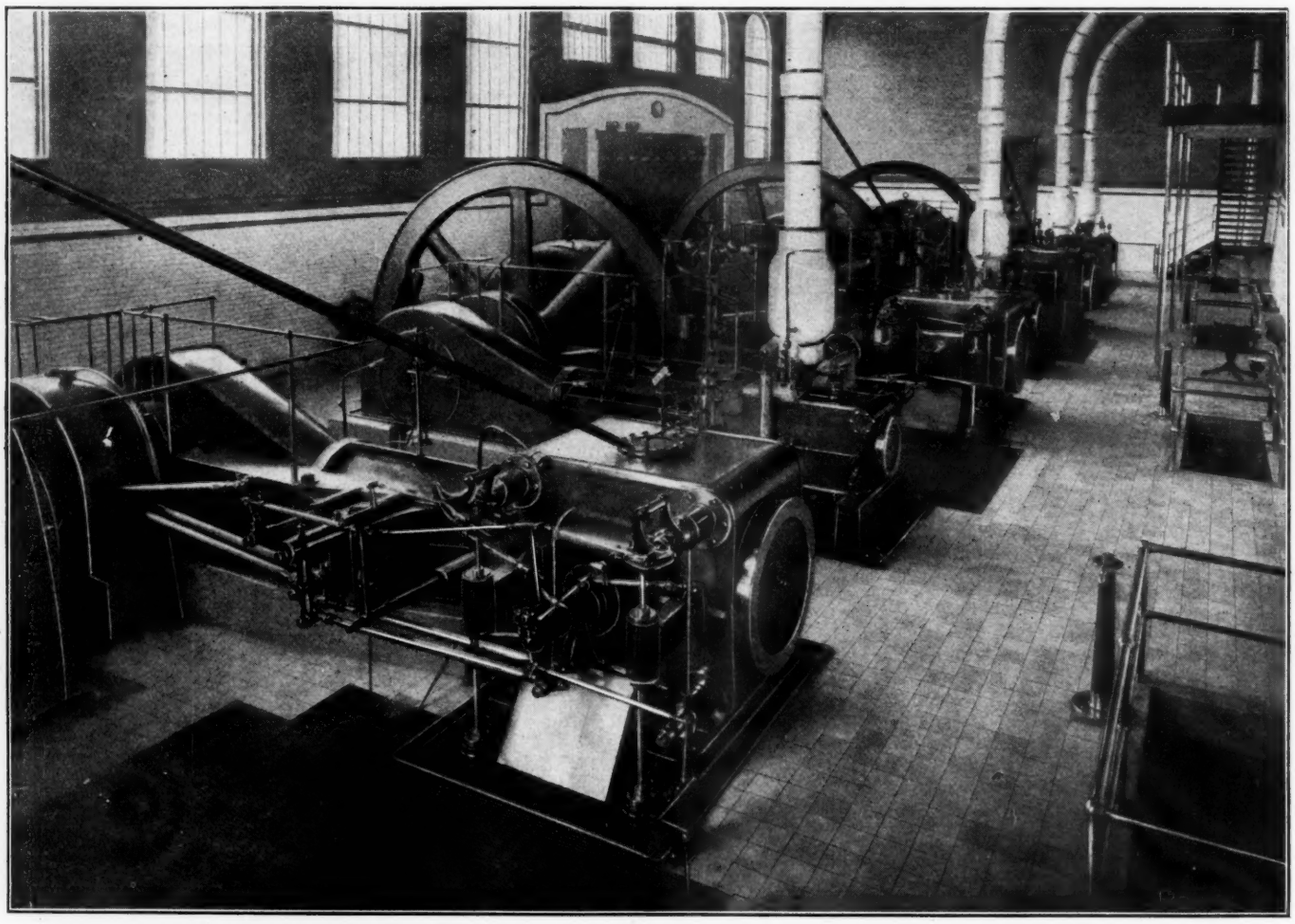


FIG. 1. ENGINE ROOM OF WISCONSIN'S CAPITOL PLANT

PLANT EQUIPMENT

The building is a one-story structure with basement in which the condensers and other steam auxiliaries are located. The engine room is approximately 40x98 ft. and the boiler room, 82x95 ft. In the latter are four Stirling water-tube boilers rated at 400 hp., and space has been reserved for four more similar units when required. Murphy smokeless furnaces, having a grate area of 80 sq.ft. each, serve the boilers. The chimney is 10 ft. in internal diameter, 250 ft. high and of radial brick. The draft is controlled by a Spencer damper regulator operating a damper in the main breeching according to the steam pressure.

Coal is first weighed on a Howe railroad-track scales and, after passing through a crusher, is delivered by a conveying system to any one of eight bunkers, each having a capacity of 200 tons. From the bunkers it is fed by gravity through Avery automatic scales and protected chutes to the magazines at the furnaces. Ashes are handled by the same conveyors used for coal, but in the reverse direction.

Feed water from the general-service mains or from the hotwell is supplied through a feed-water heater to the boilers. The feed is controlled by Liberty regulators and overpressure on the line is prevented by governors on the pumps, of which there are two. For general service there are two duplex pumps, each having a capacity of 500,000 gal. per day of 24 hr., and controlled by Fisher governors. For fire protection there are two motor-driven three-stage centrifugal pumps each of 2,500,000 gal. a day capacity against a total head of 357 ft. when running at 1200 r.p.m. A steam-driven two-stage air compressor completes the pump-room equipment.

The engine-room equipment consists of three Nordberg cross-compound condensing Corliss engines, with cylinders 14 and 28 by 36 in., operating at 120 r.p.m., each served by a Nordberg crank

and flywheel jet condenser, and one twin noncondensing engine with 14x36-in. cylinders operating at 120 r.p.m. To each of the four engines is directly connected a 250-kw. generator, making the total normal output 1000 kw. Also in the engine room are two balancer sets, each having a capacity of 12½ kw., and the main switchboard with the usual equipment of instruments and switches.

The keeping of accurate station records has been simplified, as far as possible, by installing the following curve drawing and recording devices: Four venturi meters with indicating dial and chart recorders, one on each discharge pipe from the boiler-feed pumps, one on the 3-in. high-pressure steam line in the tunnel and one on the 12-in. low-pressure steam line in the tunnel; one curve-drawing ammeter on the main busbars, recording pressure gages for steam and water, Avery automatic scales for coal, indicating thermometers for all important tem-

FIG. 2. FRONT EXTERIOR OF PLANT

perature records except flue gases, indicating draft gages, integrating wattmeters on generator leads and indicating volt and ammeters. The station is also provided with a mercury column for checking up vacuum gages, a dead-weight gage tester for pressure gages and a pair of indicators for engine and pump use.

The first boiler was fired on Mar. 16, 1910, and by the first of 1911, the plant was in regular commission and operated under the following conditions: Steam pressure at boilers, 140 lb.; steam pressure on heating main, 7 to 8 lb.; water-pressure domestic service, 90 to 100 lb.; water-pressure fire service, 150 lb.; voltage, power circuits, 230; voltage, lighting circuits, 115.

During the heating season, the engines are run simple, noncondensing, and the exhaust steam is used for heating as far as it will go. There is use for practically all of it from Oct. 1 to June 1. The accompanying tables and curves show in detail the distribution of power, heat and the relations of the various services to each other and to the weather conditions.

Fig. 5 is a typical load curve showing the variations for the 24 hr. ending at midnight, Dec. 22, 1911. The peak in the morning occurs between 10 and 11, instead of between 8 and 10 o'clock. Fig. 6 shows the monthly output in kilowatt-hours for the year 1911. The large outputs for March, April and May were due to the night sessions of the legislature. Fig. 7 is a temperature-coal curve for 1911. The fan system was not in service in the east wing during the first two or three months of the year. Fig. 8 is a temperature-coal curve for January, 1912. It shows with fair accuracy the relation of coal consumption to temperature for a building served from an independent plant. The service requires furnishing steam to the capitol for various uses throughout the entire year, and this with that required for operating the generating units necessitates operating the plant continuously whether any heating is done or not. The minimum load in the summer is not over 200 boiler horsepower, about half rating for one boiler, and it has not exceeded 300 boiler horsepower during the four warm months, June to September, inclusive. This establishes a minimum consumption of about 10,000 lb. of feed water for the plant during the nonheating period and running condensing. When heating begins and the exhaust from the engines is used for that purpose, the quantity is probably not increased and may be reduced somewhat, so that out of a total evaporation of 20,000 lb. per hr., as accounted for by the meters, from 10,000 to 12,000 lb. is probably delivered to the capitol.

Fig. 9 is a series of curves showing ash content, the heat values of the fuel and the rate of evaporation per pound of coal as fired and per pound of combustible. The feed water as recorded by the meters was discounted 10 per cent., and the coal used as recorded by the scales was increased 5 per cent. to cover any errors that might creep into the records. It may be noticed that the evaporation is higher in the warm months than in the winter. The average for the year is about 10 lb. per pound of combustible corresponding to an efficiency of boiler and grate of about 73 to 76 per cent.

Fig. 10 is a feed-water chart for the period from 6:30 a.m., Dec. 22, to 6:30

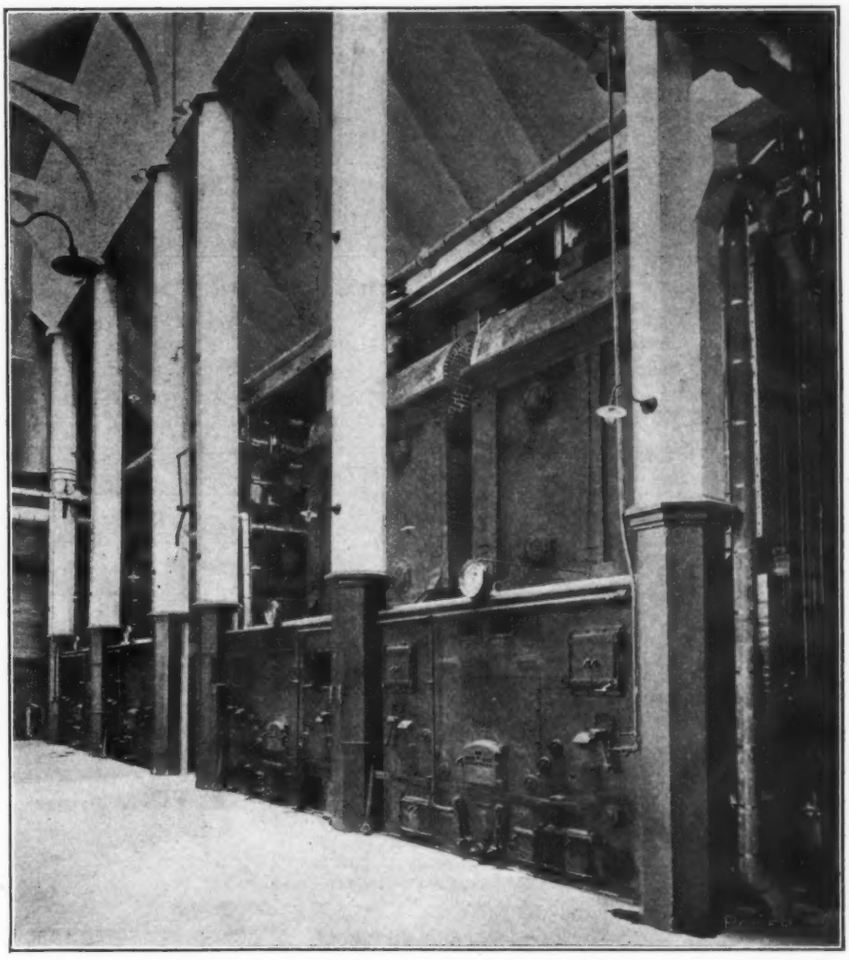


FIG. 3. EAST HALF OF BOILER ROOM

a.m., Dec. 23. The average temperature for the day as given by the weather report was 31 deg. This chart was made to show the effect on the fuel consumption of running simple and exhausting into the heating system as against running compound condensing and furnishing live steam for heating. The former method was in use when the record was started at 6:30 a.m., Dec. 22. At 9:30 a.m., a compound-condensing engine was started and it carried the load from that time until 6:30 p.m. It may be noticed that the feed-water rate increased at once from about 14,000 lb. per hr. to about 18,000 lb., and that it remained at that point until after the load was changed back to the other engine. About 500 lb. more coal per hour was required, or from 5 to 6 tons per day, although the same work was being done. The above data is a convincing argument in favor of the method under which the plant is being operated. Stated in terms of efficiency, about 12 per cent. of the fuel value is being used in the compound-condensing engine, mechanical energy alone being produced, while in the simple engine from 6 to 8 per cent. of the steam passed is converted into mechanical energy, leaving the remainder, 92 to 94 per cent., available for use in the heating system and thus securing an overall efficiency of practically 100 per cent. for this method.

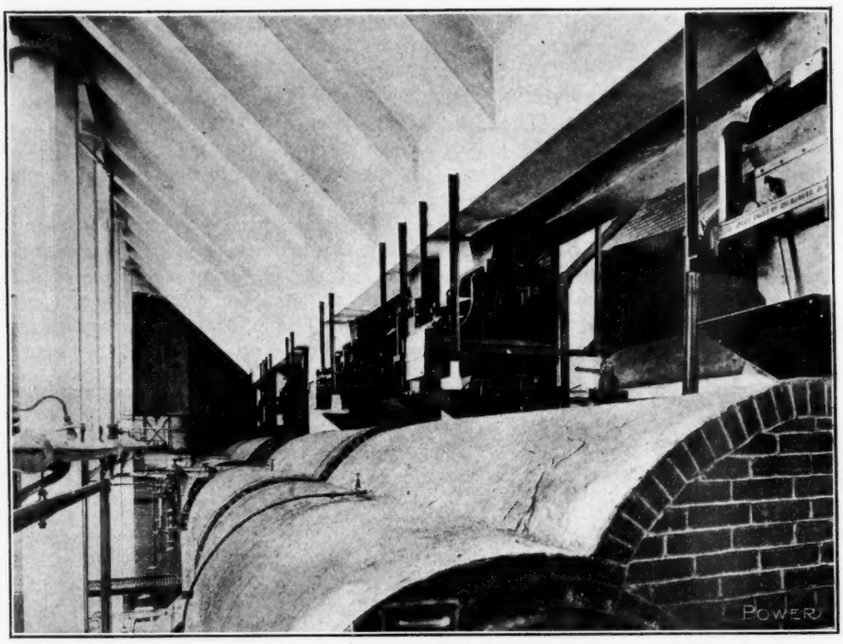


FIG. 4. AVERY AUTOMATIC SCALES ABOVE BOILERS

In Table 1 are given the more important items of the monthly records for 1911 just as they came into the office. Coal weights are from the readings of the Avery scales, kilowatt-hours from the Thompson recording wattmeters and pumpage from the displacement of pumps with a 3 per cent. allowance for slip and leakage.

The coal received at the plant from April, when the new coal contract was put into force, to the end of the year amounted to 5212.95 tons. Corrected for moisture, when it proved to be above or below the specified amount, 7.95 per cent., the tonnage was 5190.69 and cost $13,307.79. On Jan. 1 there was enough coal on hand to run until Apr. 1, so that the quantity given may be taken as a year's supply. Deductions made for excessive moisture and other penalties inflicted amounted to $245.08, which was more than enough to cover all costs of analyses and tests.

Operating expenses are given in Table 2.

TABLE 1. RECORDS FOR THE YEAR 1911

Month	Coal		Feed Water			Output		Pumpage Record	
	Monthly total, tons	Daily average tons	Monthly total lb.	Daily average	Per hour	Total kw.-hr.	Daily average	Total gallons	Daily average
January	491.05	15.85	9,880,000	318,700	13,250	74,070	2,390	1,059,000	34,200
February	459.80	16.40	8,793,000	314,000	13,900	71,280	2,545	681,000	24,300
March	512.95	16.50	9,529,000	307,400	12,800	84,720	2,730	1,176,000	38,000
April	461.20	15.35	9,187,000	306,233	12,750	73,840	2,460	891,000	29,800
May	417.45	13.45	8,809,000	284,000	11,840	71,640	2,310	1,572,000	50,700
June	398.40	13.25	8,663,000	289,000	12,040	71,980	2,400	1,487,000	49,500
July	330.90	10.94	7,162,000	231,000	9,620	60,000	1,940	617,000	20,000
August	310.40	10.00	6,698,000	216,000	9,000	62,600	2,020	835,000	27,000
September	344.15	11.45	7,891,000	263,000	10,950	58,580	1,955	1,499,000	50,000
October	442.55	14.25	9,326,000	300,900	12,500	70,100	2,260	1,195,000	36,000
November	545.75	18.20	9,929,000	368,000	15,320	61,050	2,040	2,348,000	77,200
December	562.30	18.14	10,864,000	374,000	15,590	64,780	2,090	1,914,000	61,700
Total	5276.90		106,721,000			824,640	2,262	15,274,000	41,533

TABLE 2.—OPERATING EXPENSES OF PLANT

Lubricants	
Internal cylinder oil, 49 gal. @ 75c.	$36.75
Internal cylinder oil, 496 gal. @ 43.4c.	215.27
External oil, 50 gal. @ 9c.	4.50
External oil, 202.1 gal. @ 20.4c.	41.23
Grease, 75 lb. @ 15c.	11.25
Total	$309.00
Boiler Compounds	
Six barrels, 3180 lb. @ 8c	$254.40
Waste and Rags	
No. 2 white waste, 4 bales, 459 lb. @ 10.5c.	$48.20
No. 1 white waste, 2 bales, 205 lb. @ 12.5c.	25.63
Washed rags, 100 lb. @ 4c	4.00
Total	$77.83
Unclassified	
Packings, gage glasses, paints, coke, graphite, soap, washing powder, brooms, mops, etc., estimated,	$100.00
Repairs	
Estimated 2 per cent. on $75,000 of operating equipment	$1500.00
Labor	
Chief operating engineer	$2500.00
Three assistant engineers @ $1200	3600.00
Three firemen @ $900	2700.00
One helper and steam fitter @ $900	900.00
One janitor @ $840	840.00
Six helpers @ $720	4320.00
Total	$14,860.00
Total operating expenses including coal	$30,409.02

TABLE 3.—APPORTIONMENT OF COSTS TO VARIOUS SERVICES

Item	Total cost	Steam	Electric	Other uses
Land	$30,100.00	$13,600.00	$4,400.00	$12,100.00
Tunnel	49,214.00	17,963.00	2,559.00	28,692.00
Tun. wiring	1,085.00	181.00	0.00	904.00
Chimney	16,675.00	16,675.00	0.00	0.00
Main bldgs.	123,503.00	75,003.00	38,500.00	0.00
Conv. bldg.	20,000.00	20,000.00	0.00	0.00
Boiler equipment	87,115.00	87,115.00	0.00	0.00
All piping	69,093.00	56,800,00	3,093.00	10,000.00
Gen. and trans.	76,922.00	0.00	76,922.00	0.00
Water works	28,160.00	0.00	0.00	28,160.00
Totals	$501,867.00	$286,537.00	$125,474.00	$89,856.00
Per cent	100.00	57.20	25,00	17.80
Interest and depreciation at 8%	$40,149.36	$22,922.96	$10,037.92	$7,188.48

APPORTIONMENT OF FIXED CHARGES

In the following analysis the various services are classified with reference, first, to their relative importance, and, second, to their possible competitive or noncompetitive obtainment. It is worthy of note that the furnishing of the noncompetitive services would have required

practically all of the investment, except that directly chargeable to electric service and spent for such equipment, had the latter been omitted. The services are classified as

Heating .....noncompetitive.

Tunnels for traffic as warehouse connection .....noncompetitive.

Water for fire protection..noncompetitive.

Water for general service...competitive.

Current for light and power..competitive.

In Table 3 the division was made under the various headings as set forth below:

Land—proportional to areas occupied by main buildings and accessories.

Tunnel—clear area charged to traffic as warehouse connection; remainder proportional to area occupied. There is no occasion for tunnels for either cables or pipes.

Chimney—all to steam; whether or not there were generators, a chimney is necessary.

Main buildings—proportional to cubic contents. Bunkers—considered double construction for that space.

Conveyor building—all to steam. Its only use is to handle coal.

Boiler equipment—all to steam.

Piping—proportional to estimated cost for each service.

Generators and transmission—all electric.

Waterworks—all charged to other uses.

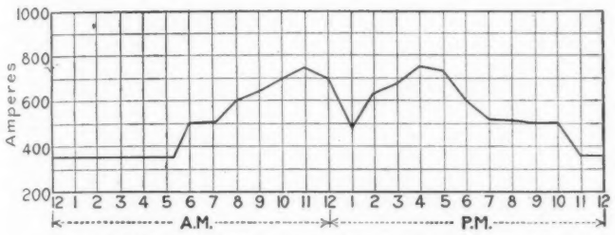


FIG. 5. TYPICAL DAILY LOAD CURVE, DECEMBER, 1911

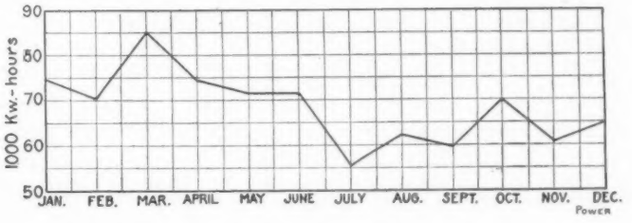


FIG. 6. MONTHLY OUTPUT IN KILOWATT-HOURS

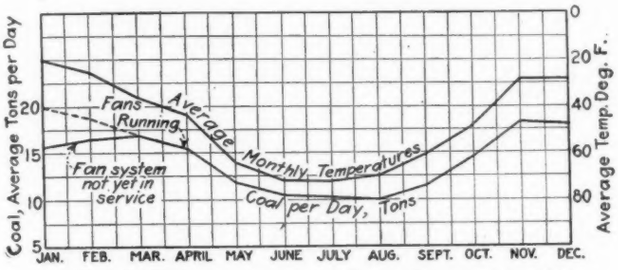

Coal, Average Tons per Day
Average Temp. Deg. F.
Fans Running
Average Monthly Temperatures
Fan system not yet in service
Coal per Day, Tons
JAN. FEB. MAR. APRIL MAY JUNE JULY AUG. SEPT. OCT. NOV. DEC.

FIG. 7. TEMPERATURE AND COAL CURVES FOR 1911

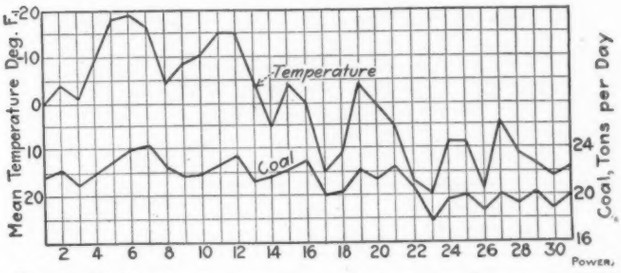


FIG. 8. TEMPERATURE AND COAL CURVES FOR JANUARY, 1912

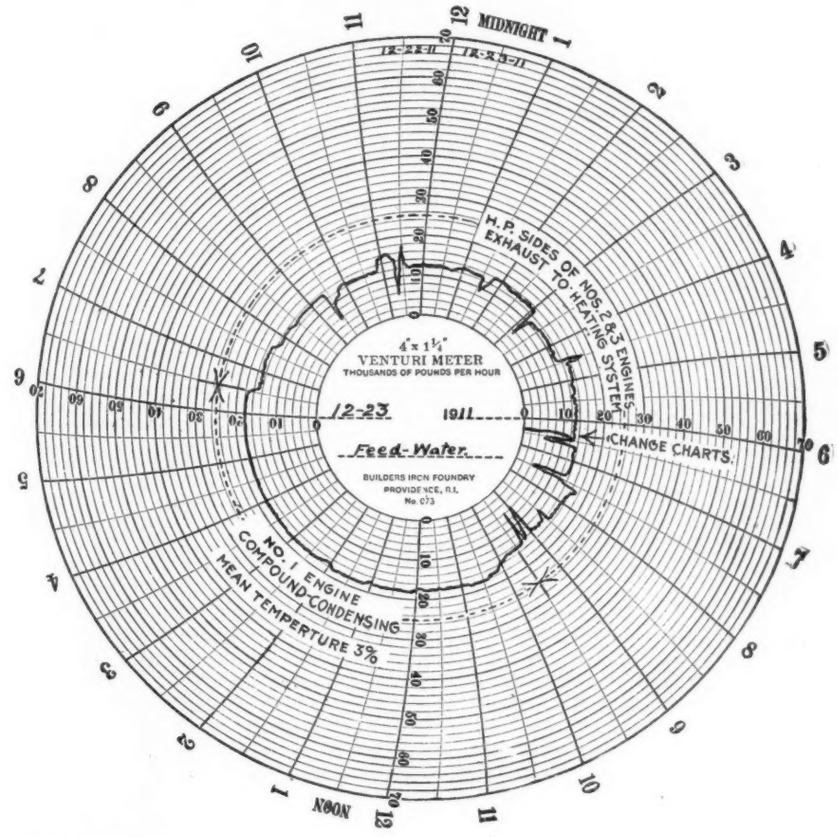


FIG. 10. FEED WATER CHART SHOWING ECONOMY OF USING EXHAUST FOR HEATING

APPORTIONMENT OF OPERATING EXPENSES

The total operating cost shown elsewhere to amount to $30,409.02 for the

TABLE 4. APPORTIONMENT OF OPERATING EXPENSES

Item	Total	Steam	Electric
Fuel	$13,307.79	$9,634.84	$3,672.95
Lubricants	309.00	109.00	200.00
Boiler compounds	254.00	200.00	54.00
Waste, rags	77.83	27.83	50.00
Unclassified	100.00	50.00	50.00
Repairs, estimated	1,500.00	1,000.00	500.00
Labor	14,860.00	12,000.00	2,860.00
Totals	$30,408.62	$23,021.67	$7,386.95
Per cent	100.00	75.70	24.30

TABLE 5.—GENERAL SUMMARY

Item	Steam	Electric	Other uses
Operating	$23,021.67	$7,386.95	0.00
Int. and dep.	22,922.96	10,037.92	$7,188.48
Totals	$45,944.63	$17,424.87	$7,188.48

TABLE 6.—UNIT COSTS

Item	Total cost	Total output	Unit cost
Electricity	$17,424.87	824,640 kw.-hr.	2.11c per kw.-hr.
Steam	$45,944.63	2,784,000 boiler hp.	1.65c per boiler hp. hr.
Cost per kw.-hr. exclusive of interest and depreciation			0.899c
Cost per boiler hp.-hr. exclusive of interest and depreciation			0.827c
Cost to evaporate 1000 lb. water exclusive of interest and depreciation			23.97c

year ending Dec. 31, 1911, is apportioned in Table 4 between steam and electric. No separation has been attempted as between steam and pumping operations and other uses. A rational division would be almost impossible to obtain and of no value. All of the pumps are steam-driven except those for tunnel drainage and fire pumps.

Table 5 gives a general summary of the apportionment and Table 6 unit costs.

As the plant is being operated at considerably less than half capacity for steam and at only about one-fourth capacity for current the overhead charges will be reduced greatly per unit of output as the plant takes on the load for which it was designed. With the completion of the capitol and the construction of the other buildings contemplated, a much better showing can be made.

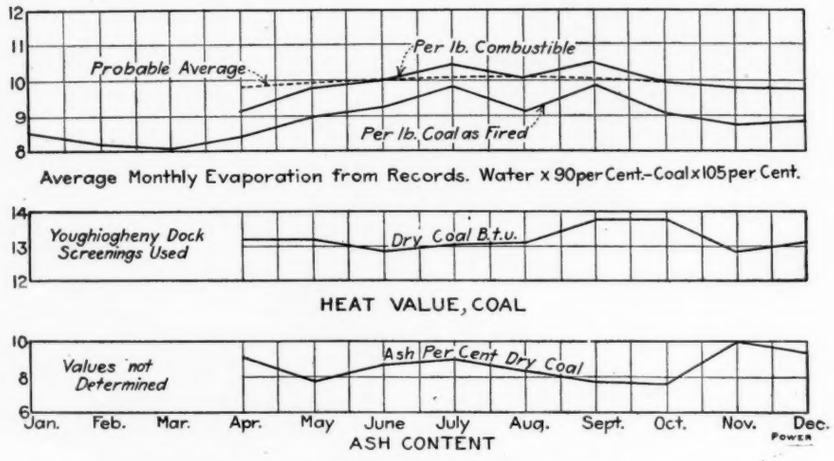

Probable Average
Per lb. Combustible
Per lb. Coal as Fired
Average Monthly Evaporation from Records. Water x 90 per Cent.-Coal x 105 per Cent.
Youghiogheny Dock Screenings Used
Dry Coal B.t.u.
HEAT VALUE, COAL
Values not Determined
Ash Per Cent Dry Coal
Jan. Feb. Mar. Apr. May June July Aug. Sept. Oct. Nov. Dec.
ASH CONTENT
POWER

FIG. 9. COAL VALUES AND EVAPORATION

Replaced Stokers with Shaking Grates

BY V. L. BOWLES

Although contrary to present-day practice and the cause of much criticism, the change from stoker equipment to hand firing has borne out the judgment of the writer to such extent in the plant he operates that the balance of the stoker equipment will be replaced by hand-fired furnaces.

The reasons for the change were high cost of upkeep, large amount of combustible in the ash, unreliability during long and hard periods of operation and the fact that no extra labor would be required.

The stoker referred to was one of the most modern of the toboggan-grate type. Due to the layout of the plant, proper coal-handling apparatus could not be installed with any degree of success; therefore all coal was weighed and handled by hand.

There are three boilers of the vertical water-tube type of 100, 250 and 350 hp. respectively, furnishing steam for a power plant and about 60,000 sq.ft. of direct radiation, all condensation being returned to the boilers during the winter with sufficient raw water to prevent pitting.

During severe winter weather the two larger boilers will handle the work without excessive overload, but there is no reserve in case of breakage to either stoker. This feature would, of course, be eliminated by a hand-fired installation; the quality of the feed water is such that the boilers could be operated for a long period, if necessary, without cleaning.

All three boilers are equipped with dutch-oven furnaces, the smaller one being hand-fired.

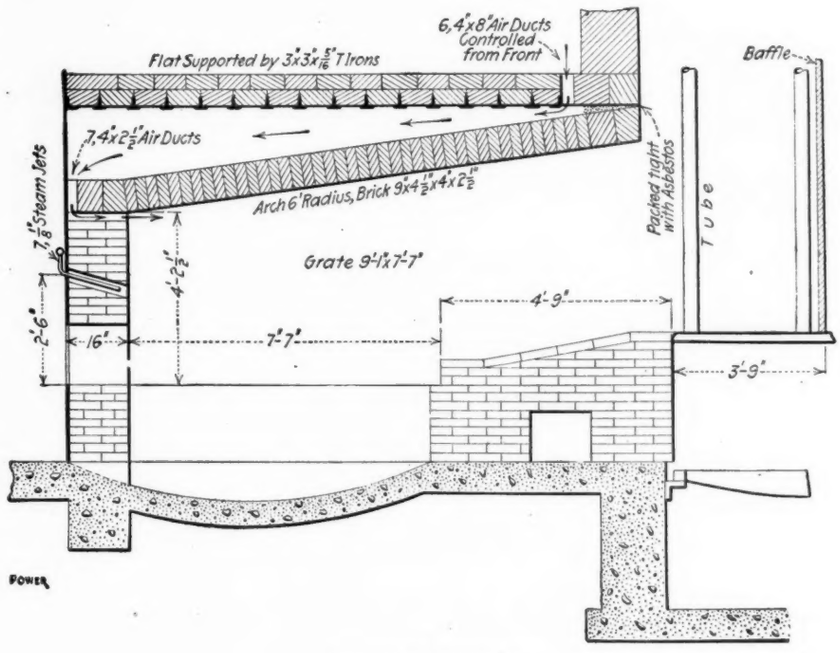
After long and close observation of the different methods of firing, and comparison of results relative to economy, reliability, upkeep and labor, the stoker on the largest boiler was discarded and shaking grates installed.

The accompanying longitudinal section of the new furnace, with principal dimensions, will show the general construction.

Steam jets were an advantage in maintaining a higher and more uniform temperature and eliminating smoke, which has been cut down 75 per cent. Although the new furnace has been in use but a comparatively short time, a saving of approximately 4 per cent. in coal of the same grade as formerly used with the stoker, with about the same load, a much cleaner ash, more uniform steam pressure and less labor for the fireman resulted. The last two statements may appear impossible, nevertheless they are true, due partly to local conditions.

There is no question but that the upkeep will be reduced at least 80 per cent. The accessibility is also much greater. Sudden overloads are handled with just as much ease and with less variation of steam pressure than formerly.

Three electric pyrometer couples are used for obtaining furnace temperatures, one in the first pass, 9 ft. above the drum, one in the second or downward pass, 9 ft. from the lower drum and one at the base of the stack.



DESIGN OF FURNACE

The temperature at the first couple will vary from 1400 to 1900 deg.; at the second, from 675 to 877 deg., and the stack temperature will vary from 550 to 675 deg. with a draft over the fire of from 0.2 to 0.35 in., depending on the load.

The CO_2 runs almost exactly the same as with the former stoker equipment.

It has been found that close attention to the temperatures at the three different points will net better resuls than equal attention to the stack gases, although nearly the same methods are employed in arriving at the best values. Different grades of coal will be tried out in the near future with no doubt still better results.

Burning Natural Gas under Boilers

By Leon B. Lent

Notes on the use of natural gas as fuel, together with data compiled from different sources, showing the results of actual tests.

In the matter of burning natural gas, the experience of several authorities shows agreement in several points, but there is a radical difference of opinion on others. Accurate information is somewhat meager and in most cases it is difficult to reduce results to a comparative basis.

Burners

All types of burners seem to show about the same efficiency. The majority are of the inside-mixing type; that is, the gas is delivered under a pressure of from 4 to 8 oz. of water (sometimes as high as 30) and in a direction somewhat tangential to the axis of the burner. This produces a helical swirl of the gas and the induced air and is conducive to a more thorough mixture. The proportions of gas and air should be under accurate control. Better results are obtained (as with oil fuel) by using several burners of smaller size than one or more burners of larger size.

Furnaces

According to J. Rowland Brown (*Transactions*, A. S. M. E., Vol. 27, page 188), the greatest difference in efficiency between two different boilers lies in the location of the brick checkerwork in the furnace, the best results being secured with this about 3 ft. beyond the burner. This checkerwork acts as an igniter for the gases with a second checkerwork 3 ft. further on to break up the flame passing through the first. Mr. Brown also contends that the overhead arch does not make as much difference in a gas furnace as in a coal furnace, the checkerwork being the important factor.

The same conditions are necessary for the efficient combination of gas as for any other fuel; namely, proper proportions of gas and air well mixed and burned under the highest possible furnace temperature. The latter means that the combustion chamber should be surrounded with refractory material capable of assuming and maintaining a sufficiently high temperature (in good practice a high state of incandescence).

The furnace should be of sufficient volume if possible to allow complete combustion before it is quenched by contact with the cooler heat-absorbing surface of the boiler proper.

While some tests show practically equivalent performances when gas is burned with a blue flame and with a white or straw-colored flame, it is generally conceded that the blue flame indicates more efficient combustion, a higher temperature and a superior performance, other things being equal.

Boiler Performance

Assuming natural gas to have a heat value of 1000 B.t.u. per cubic foot and a boiler horsepower to be equivalent to

$$34.5 \times 970.4 = 33478.8 \text{ B.t.u.}$$

absorbed per hour, the gas consumption for various combined boiler and furnace

TABLE 1. EVAPORATION AND GAS CONSUMPTION AT DIFFERENT BOILER EFFICIENCIES

Combined boiler and furnace efficiency, per cent.	Cu.ft. of gas per b.hp.; heat value = 1000 B.t.u. per cu.ft. of gas $\left(\frac{34.5\times970.4\times100}{1000\times\text{Efficiency}}\right)$	Evaporation from and at 212 deg. per cu.ft. of gas $\left(\frac{1000\times\text{Efficiency}}{970.4\times100}\right)$	Evaporation from and at 212 deg. per lb. of gas with heat value of 19,510 B.t.u. per lb. of gas $\left(\frac{19{,}510\times\text{Efficiency}}{970.4\times100}\right)$
100	33.5	1.030	20.1
90	37.2	0.926	18.1
85	39.4	0.875	17.1
84	39.8	0.865	16.9
83	40.3	0.854	16.7
82	40.8	0.844	16.48
81	41.3	0.833	16.28
80	41.8	0.823	16.07
79	42.3	0.813	15.87
78	42.8	0.803	15.67
77	43.4	0.7925	15.47
76	44.0	0.782	15.27
75	44.6	0.772	15.07
74	45.2	0.762	14.87
73	45.8	0.752	14.67
72	46.4	0.741	14.47
71	47.1	0.731	14.27
70	47.8	0.721	14.06
69	48.5	0.711	13.86
68	49.3	0.700	13.68
67	49.9	0.690	13.47
66	50.7	0.679	13.26
65	51.5	0.669	13.06

TABLE 2. RESULTS OF BOILER TESTS WITH GAS FUEL

No. of test	Type of boiler	Steam pressure, pounds	Feed water, temp. deg. F.	Type and number of burners	Gas pressure at burner, ounces	Boiler rating, hp.	Boiler hp., developed	Cu.ft. of gas per boiler hp.	Combined boiler and furnace efficiency	Test conducted by
1	Cook water tube	126.5	33	10 Gwynn	3.84	250	254.5	48.3	69.2	J. M. Witham
2	Cook water-tube	122	33.6	10 Kirkwood	4.5	250	253.7	48.0	69.7	J. M. Witham
3	6 Cook water-tube	112	172	60 Kirkwood	4.0	Each 250	247	41.0	81.6	J. M. Witham (with white flame)
4	6 Cook water-tube	88	178	60-6-in. Kirkwood	4.0	Each 250	213.1	41.0	8.16	J. M. Witham (with blue flame)
5	6 Cook water-tube	116	144	60–6–in. Kirkwood	6.0	250	296.6	45.2	74.0	J. M. Witham (white flame)
6	6 Cook water-tube	99	151	60 Kirkwood	6.0	250	271.1	42.1	79.0	J. M. Witham (blue flame)
7	6 Cook water-tube	128	157	42 Kirkwood	8.0	250	225.4	44.2	75.7	J. M. Witham (white flame)
8	6 Cook water-tube	128	184	46 Kirkwood	8.0	250	227.6	47.2	70.9	J. M. Witham (blue flame)
9	6 Cook water-tube	120.1	151.2	Kirkwood	6.9	250	273	44.9	74.6	J. M. Witham
10	.....	132.7	185	Kirkwood	6.4	250	252	40.96	81.7	J. M. Witham
11	Heine water-tube	145.8	40.9	6 Kirkwood	.....	200	154.9	48.6	68.8	G. I. Bouton
12	Heine water-tube	144.1	61.1	6 Kline	.....	200	217.8	43.0	77.7	G. I. Bouton
13	Heine water-tube	149.6	74	6 Kirkwood	.....	200	258.0	40.51	82.4	(Supplemental supply of preheated air used.) Bouton and Witham
14	Horizontal Cahall	95.0	46.5	6 Merrill	4.8	302	340.2	44.5	75.0	J. Rowland Brown
15	Horizontal Cahall	85.7	53.5	6 Merrill	7.0 to 30.0	302	260.0	37.4	89.2	J. Rowland Brown
16	Horizontal Cahall	87.2	54.3	12 Gwynn	16.0	302	532.8	44.2	75.7	J. Rowland Brown
17	Horizontal return tubular	90.0	176.5	Klein	2.21	100	55.0	48.8	68.7	E. G. Bailey
18	Horizontal return tubular	88.0	164.5	Klein	2.58	100	64.8	46.6	71.7	E. G. Bailey

efficiencies will be as shown in Table 1.

In a boiler fired with natural gas and having a properly designed furnace, everyday practice should show a combined efficiency of 70 per cent., according to one authority, and this seems quite reasonable. There should be no large amount of excess air, the combustion temperatures should be high, the furnace doors may remain closed and but little soot should be formed to deposit on the tubes.

The principal qualities derived from several boiler tests are given in Table 2. The figures given do not represent the best practice in all cases nor do all the figures admit of comparison on the same basis, since the boilers tested are different types, run under different conditions, and some with obvious inefficiency. The calorific value of the gas is not given for all the tests; in tests 1 to 10 inclusive, it was 1098 B.t.u. at 60 deg. F. and 29.92 in. barometer, and in tests 17 and 18 the value was 1199 B.t.u. at 32 deg. F. and 29.92 in. barometer.

Height of Chimney

The chimney height necessary to supply draft for natural gas is that necessary to overcome the frictional resistance of the gas passages from the furnace to the top of the chimney and seldom exceeds 0.20 in. of water unless an economizer is in the circuit. A stack 80 ft. high should meet any demand for intensity of draft, although local conditions may make it necessary to discharge the furnace gases at a much greater height. Dampers should be used in any case, for excessive draft will induce an excessive air supply and so decrease the efficiency.

Daly and Doolin Discuss Riveted Joints

By Danny Hogan

"Did ye see th' discussion goin' on be th' paaper," said Daly, "about th' pitch of rivets in b'iler seams? Sure 'tis a most insthructive set af articles, an' ivery b'iler maker should be radin' up on th' subject. But th' multichude of writers splashes a pint av good ink showin' th' importance av th' proper pitch an' how to find th' efficiency av th' seam, an' all that. Thin they sthop."

"Oi read thim," replied Doolin, "an' agree wid ye that they were hoighly insthructive."

"Engineers should pay more attintion t' b'iler design, Doolin, for th' funny part av it all is, thim high-brow fellers niver git anny further than analyzin' th' efficiency av th' seam."

"Oi admit 'tis important," said Daly, "but 'tis like speakin' only av th' pitcher in a baseball game an' forgettin' intoirely th' other eight min. A riveted seam is a chain wid siveral links, an' 'tis important t' know th' strength av each link, wan av which, be th' way, is th' straight pitch. And, moind ye, wan must choose a diameter for th' rivet hole for a certain thickness av plate an' not run away wid th' idea that anny diameter av rivet will do fer anny thickness av plate. Nayther should th' size av th' rivet head, th' height an' diameter at th' plate, an' th' shape be disraymimbered."

"Do ye think a button head, steeple, cone or conoidal head th' sthrongest?" asked Doolin, as he filled his "T. D."

"Th' heads must hold th' plates t'gither an' sthand up for calkin' th' seam," answered Daly, as he handed Doolin a brimstone match. "Always raymimber a rivet in shear is loike a bolt in tinsion, an' th' head has t' be considered, for all th' pitch writers neglicted this p'int.

"Thim chaps, Doolin, niver mintion whether th' rivet holes was punched full-soized, punched small an' reamed out in place, drilled with th' plates flat or drilled after rollin' th' plate up. Now all this constitoots a link in th' value ov a j'int whether th' ingineers think wid me or not, an' it makes th' difference in th' price of a b'iler as well. If ye are lookin' fer a cheap b'iler, just ignore th' question av makin' th' rivet holes, but if it's a good one ye want, have th' holes drilled out full-sized afther th' plate is rolled up, loike is done wid th' 16-ft. Scotch marines.

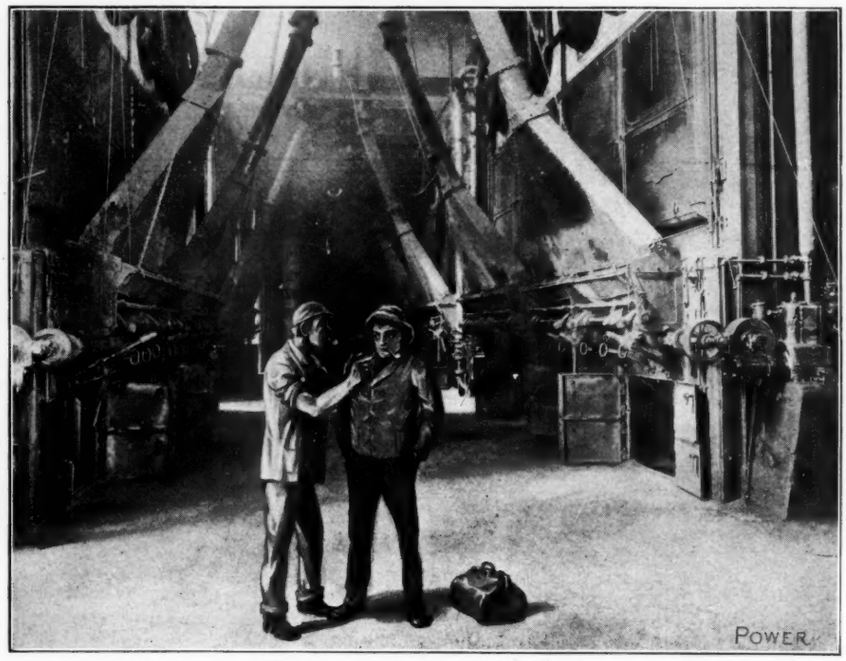
"Ye may know, Doolin, that English tests shows 18 per cent. loss in tinacity be reason ov th' punchin' ov 3/8-in. plates. Av coorse, th' higher in phosphorus th' metal, th' greater Oi believe would be th' loss. Consider, thin, this detail in findin' th' efficiency av th' rivet pitch. Say ye had a 70 per cint. joint wid holes punched full size, an' say th' mill test showed carbon 9.3 per cint., phosphorus 0.06 per cint., an' sulphur 0.05 per cint.—th' b'iler made wid a lap seam, an' 20 years old at that—what efficiency would ye allow?"

Here Doolin choked slightly as some of the sulphur fumes were inhaled. When he recovered his breath, Daly said: "Goin' further down th' pike, Doolin, let's look into th' mather av whether th' seam is a true arc av th' circle. Ye know th' usual flat spot on th' sheet is along th' longitudinal seam. Now tell me, is th' finished seam truly cylindrical or not? If not, thin flexure is prisint an' to be considered.

"There's siveral more points. Were th' rivets driven be hand, snapped by th' gun or be th' bull? Were they overheated an' carbonized in a coal fire or properly heated in a gas or oil furnace by a lad what was onto his job? Do they fill th' hole solidly from ind to ind or only at th' inds? If not, in th' latter case, th' shear diameter is much less an' that's a factor in yer analysis of th' p'int.

"Of course, Doolin, th' bull-drove rivet is th' best, anny an' all things considered, an' for why? Th' squeeze is exerted on sthraight loines an' th' prissure is therefore concintric an' not eccintric to th' rivet. The force exerted in drivin' is that powerful it is not nadeful to heat th' rivet as hot as whin workin' be other methods, an' this avoids burned rivets.

"Again, th' best b'iler shops are now usin' only 1/32-in. clearance instid ov 1/16 in.. for th' rayson they can use a dull-



"Th' Funny Part Av It, Doolin, Is That Thim High-brow Fellers Niver Git Anny Further Than Analyzin' th' th' Efficiency av th' Seam"

red rivet wid th' bull, when th' gun would need one near white hot, an' th' latter would, ov coorse, swell in diameter. It folers, ther'fore, that in th' first case th' hole is complately filled as there is less clearance to fill.

"Goin' back t' th' efficiency matter again, Doolin, let me obsarve that th' ink-slingers say nothin' av th' j'int failin' be crushin' in front av th' rivet at th' edge of th' plate. 'Tis just as much a matther t' be considered as that av th' sthraight pitch. They say nothin' nayther av th' back or distance bechune th' rows ov rivets. But at wan glance ye can see th' importance av analyzin' these features, as they are a link in th' chain an' an ilimint in th' question of safety.

"It's glad Oi am, Doolin, to note ye are readin' up on b'iler consthruction, an' Oi advance these few p'ints fer yer careful consideration, as th' politicians say. Whin ye land that job as examiner at th' city hall, if ye iver do, bear this dope in moind for questions t' put t' th' laads. Sure, they needs t' know all about th' subject as well as one part ov it.

"And now get onto yer job," concluded Daly, "fer Oi sint fer ye to replace some tubes in No. 4 b'iler an' Oi suppose ye'll be chargin' up th' time spint in discoorsin' av riveted seams."

"Only sthraight toime," answered Doolin, "fer 'tis daywork."

No-If Engine Stop

This engine stop depends upon gravity and a suspended weight which is released when the engine exceeds its safe speed limit.

Fig. 1 shows the speed-limit band *A* attached to the shaft. It is held in position on the blocks *B* by the bolt *C*. A steel spring 1¼ in. wide is held by these blocks, and the band and spring hold the adjusting blocks *E*. *F* is a weight fastened to the spring *D*. *G* is a steel lip contact. The spring is loose on the pin *H*, which prevents lateral motion.

Fig. 2 shows the location of the quick-closing valve. Fig. 3 shows the stop applied to a quick-closing valve. *A* is an adjustable arm, and centrifugal force, due to overspeed, causes the spring of the speed limit to elongate until it strikes the latch, releasing the vertical latch holding the rod *C*. This rod sustains the weight *F* suspended by a cable running through a ½-in. conduit *E* and over roller-bearing pulleys.

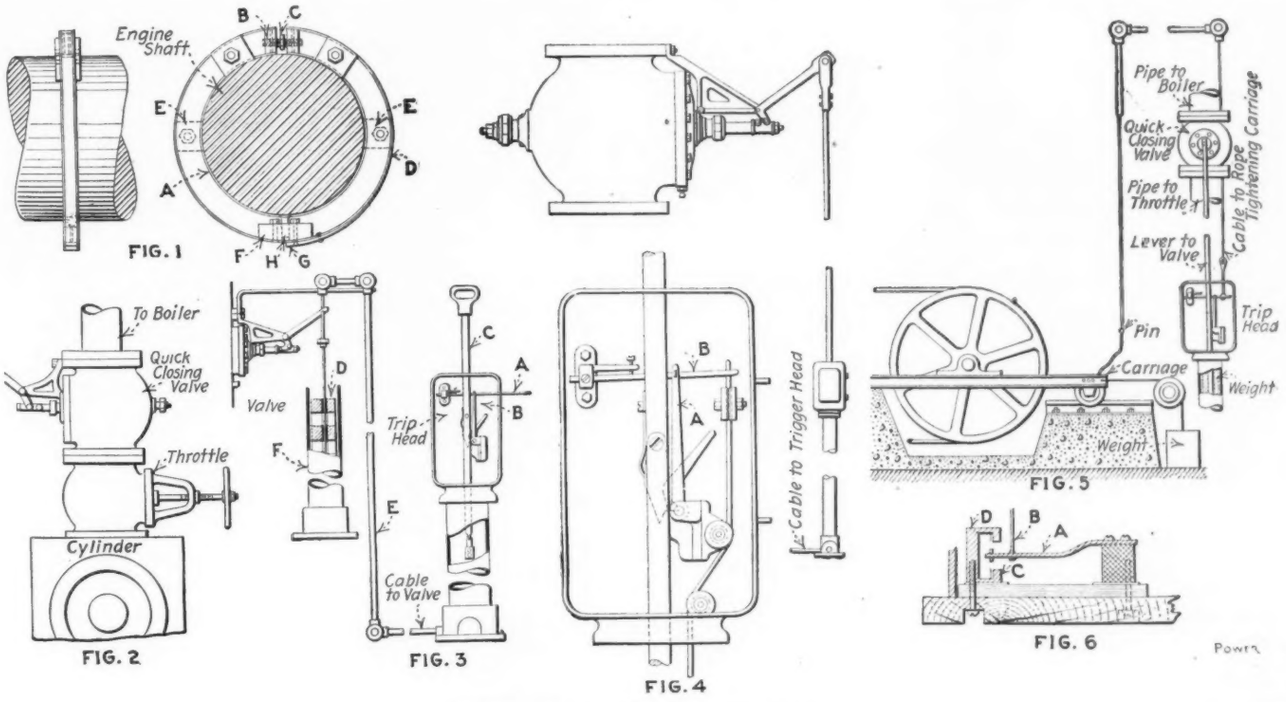
While the engine is running, the stop is operative. Should the stop go out of commission, the engine will stop, as the weight will close the throttle. To put the stop in commission, close the throttle and open the quick-closing valve by lifting the rod *C*. This opens the valve and relatches *B*. The engine cannot be run until this is done.

Another application of the stop is shown in Fig. 4. The amount of weight used to close the valve has no effect upon the latch *A*. The armature *B* must be pulled down to release the weight. When stopping is done from a distance, the armature is actuated by a magnet.

Fig. 5 shows a method of protecting an engine from running away when the American type of rope drive is used. Should the sheave crush, weight drop off, or the carriage work loose, the cable pulls the tripping latch and the throttle valve is closed before the rope can destroy the governor, thus saving the engine.

An electrical telltale device for applying to large belts and ropes is shown in Fig. 6. The parts are mounted on a slate base, the spring is held out of contact by a fine copper wire *B* stretched in front of the belt or rope. When the rope strands or belt tears, the wire is broken, contact is made at *C* and the alarm is continuous. Should the telltale be repaired with wire or string too strong to be broken by a rope strand, the spring *A* is pulled and makes contact at *D*, in which case the alarm is intermittent.

This engine stop is manufactured by the Metallic Packing & Manufacturing Co., Elyria, Ohio.



DETAILS OF THE NO-IF ENGINE STOP

Bursting Tube in Water Tube Boiler

BY A. W. REENE

On Aug. 21 a tube burst in the lower row of a water-tube boiler in the plant of the Dixie Portland Cement Co., Richard City, Tenn., instantly killing a fireman who was enveloped by the terrific volume of steam and water which came out of the furnace doors.

This tube ruptured at a point just in front of the bridge-wall. The opening was about 13 in. long, and spread out almost flat at the center of the rupture, which gave two openings for the escape of the steam and water nearly the full size of the tube. The boiler pressure was 175 lb. at the time of the accident.

An examination of the tube showed it had elongated 5/8 in. and was loose in both tube headers. A number of adjacent tubes were distorted, but were tight in the headers.

The tube was lap welded and of standard thickness. It split on the side opposite the weld. The boiler to which this accident occurred is fired with an underfeed stoker equipped with mechanical draft, the pressure being about 5 in. The fires were very heavy and the coal of a low grade. It is, therefore, quite possible that a hole was blown through the bed of fuel, allowing a jet of flame to impinge against the tube with such intensity as to overheat the tube and allow the pressure to do the rest.

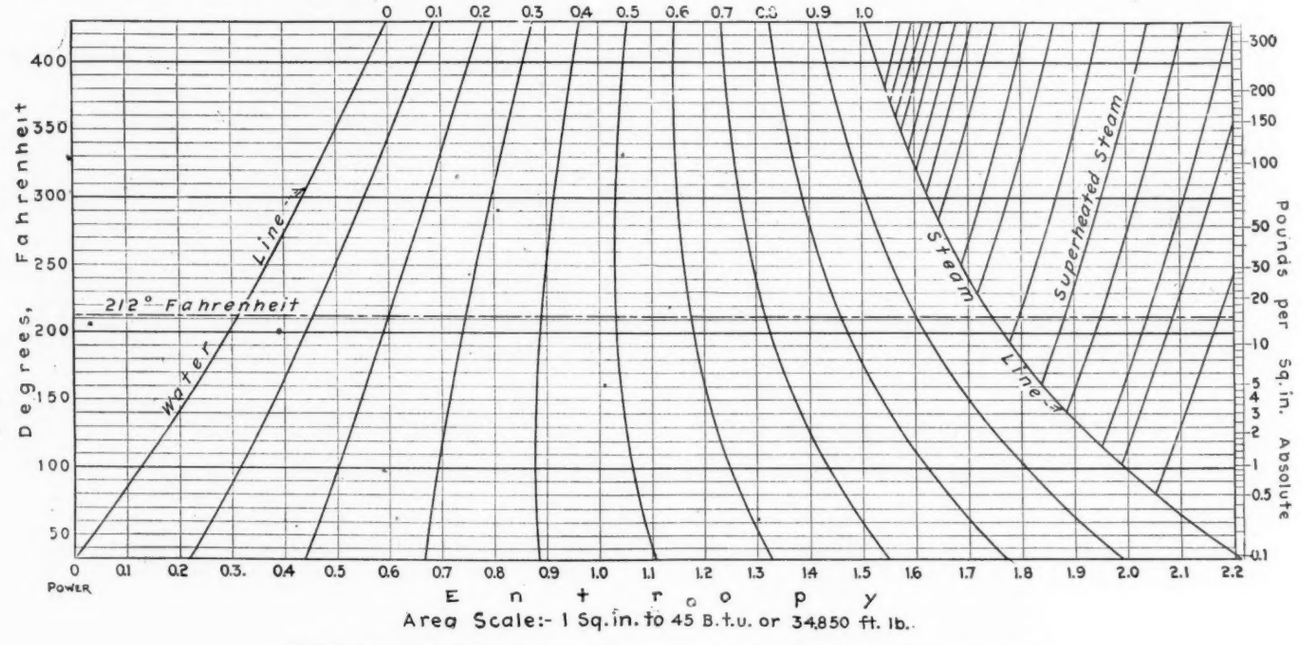
Simplicity of the Entropy Chart

By W. J. Crawford

The explanation of entropy ordinarily is perhaps rather confusing, but the use of the chart is simplicity itself. A number of steam problems are worked out with the chart. Try them and see how easy they are.

That the entropy-temperature chart is a wonderfully mysterious affair, requiring an immense amount of study to use it is a widespread idea among engineers and draftsmen. Those who have no trouble with the ordinary indicator diagram avoid the chart, because the textbooks, by their method of approaching the subject, have frightened them off. As a matter of fact, using the chart is simple and many have been exceedingly surprised when shown how easy it is to work out problems by its aid.

A general entropy-temperature chart is reproduced herewith and should be referred to, from time to time, as the explanation proceeds. For simplicity the volume and heat-unit lines found on the latest charts have been omitted. These lines are not necessary, but make using the chart more convenient. Fig. 1 represents the chart in its simplest form.



ENTROPY CHART WITH VOLUME AND B.T.U. LINES OMITTED FOR SIMPLICITY

The vertical line on the left indicates temperatures in Fahrenheit degrees and that on the right pressures in pounds per square inch, absolute, corresponding to these temperatures. The horizontal line represents entropy. The chart contains two main curves, *AB* and *CD*. The former is called the water line and *CD* the saturated steam line, or, more shortly, the steam line. The water line is nearly straight and the steam line is more curved. By tracing these two lines on the general diagram, it will be possible to work out many important examples.

EXAMPLE 1. A pound of water at 212 deg. F. is heated to 300 deg. F., and then converted into dry saturated steam at that temperature; it is then expanded adiabatically* to 212 deg. F. Find its dryness after the expansion.

SOLUTION. See Fig. 2. Mark the point *A* on the water line at 212 deg. F., go up the water line to *B* opposite the 300 deg. F. mark. From *B* go horizontally across to the steam line at *C*. From *C* draw a vertical line downward to meet the horizontal line through *A*, at *D*. Then the dryness after the expansion is given by the ratio $\frac{AD}{AE}$. Measure *AD* and *AE* carefully with a rule reading to small decimal parts of an inch. Then dividing *AD* by *AE* will give 0.91. Therefore the percentage of dryness is 91, that is to say, of the one pound of steam originally dry and saturated at 300 deg. F., only 0.91 lb. is steam after the adiabatic expansion to 212 deg. F., and consequently 0.09 lb. is water.

The steam is dry and saturated at 300 deg. F., so it is necessary to go to the point *C* on the saturated steam line, before beginning the expansion. As the expansion is adiabatic, a vertical line must be drawn to represent it on the chart.

EXAMPLE 2. A pound of water at 160 deg. F. is heated to 300 deg. F.; it is then converted into steam 80 per cent. dry at that temperature, and then expanded adiabatically to 212 deg. F. Find its dryness after the expansion.

SOLUTION. See Fig. 3. Mark the point *A* on the water line opposite 160 deg. F. Go up the water line to *B*, opposite 300 deg. F. Go along horizontally from *B*

*Adiabatic expansion is expansion of a fluid when no heat leaves it or is admitted to it during the expansion. This is approximately realized in the case of steam expanding in a well-lagged cylinder.

to C, which is $\frac{8}{10}$ the length of BK, the distance between the water and steam lines. The point C is easily found because sloping lines are drawn downward on the general chart, at each tenth between the water and steam lines. From C draw a vertical line downward to meet the horizontal line through E, drawn opposite the 212 deg. F. mark. EF is the length between the water and steam lines. Then the dryness fraction after expansion is found by dividing ED by EF. The answer will be $\frac{3}{4}$ or 0.75, and the dryness is therefore 75 per cent. By using the sloping lines labeled in tenths between the water and steam lines on the general chart the dryness could easily have been estimated after dropping down vertically to the 212 deg. line.

If the wetness after the expansion had been required, it could have been found by dividing DF by EF, which would give 0.25, or 25 per cent.

EXAMPLE 3. A pound of feed water at 175 deg. F. is heated and converted into steam at a pressure of 100 lb. per sq.in., absolute, or 85 lb. gage. The steam is wet. After it is formed it is expanded adiabatically to a pressure of 30 lb., absolute, or 15 lb. gage, when it is found to be 60 per cent. dry. Find the original dryness of the steam.

SOLUTION. See Fig. 4. Mark point A on the water line corresponding to 175 deg. F. Go up the water line to B, opposite the 100-lb. mark. Draw a horizontal line from B and opposite the 30-lb. mark draw a horizontal line DH between the water and steam lines. From D locate E at 0.6 of the length DH, to indicate a dryness of 60 per cent. after expansion. From E draw a vertical line upward to meet the top horizontal line at F. Then the original dryness of the steam formed at a pressure of 100 lb. is given by the ratio of BF to BK, which equals 0.59, or 59 per cent.

SUPERHEATED STEAM

To the right of the steam line on the general chart, sloping lines going upward may be noticed. These are the superheated steam lines.

EXAMPLE 4. One pound of steam is dry and saturated at 212 deg. F. It is then superheated to 300 deg. F., and then expanded adiabatically to 160 deg. F. Find its dryness after the expansion.

SOLUTION. See Fig. 5. Mark point A on the steam line corresponding to 1 lb. of steam dry and saturated at 212 deg. F. Go up the superheated-steam line to B, opposite 300 deg. F. From B draw a vertical line downward to meet the horizontal line drawn between the water and steam lines opposite 160 deg. F., at C. Then the dryness after the expansion is $DC \div DE = 0.98 = 98$ per cent.

EXAMPLE 5. One pound of dry saturated steam at 212 deg. F. is superheated and then expanded adiabatically to 160 deg. F. After the expansion it is again dry and saturated. How many degrees of superheat have been given to it?

SOLUTION. See Fig. 6. Mark A on the steam line corresponding to 1 lb. of steam dry and saturated at 212 deg. F. Mark B on the steam line corresponding to 1 lb. of steam dry and saturated at 160 deg. F. From B draw a vertical line upward, meeting at C the superheated steam line from A. Read the temperature opposite C. Then this temperature above 212 deg. F. is the number of degrees of superheat given the steam, or

$$335 - 212 = 123 \text{ deg. F.}$$

EXAMPLE 6. One pound of dry saturated steam at 212 deg. F. is superheated to 390 deg. F. It is then expanded adiabatically to 300 deg. F. Find the number of degrees of superheat it has after the expansion.

SOLUTION. See Fig. 7. Mark A corresponding to 1 lb. of dry saturated steam at 212 deg. F. Go up the superheated-steam line to B, opposite 390 deg. F. Draw a vertical line downward to C, opposite 300 deg. F. Through C draw a superheated-steam line CD (if there is such a line passing through C, use it; if not, draw a superheated-steam line as nearly parallel as possible to the superheated lines on either side of the point). Then D is the point on the steam line where the superheated-steam line C meets it. Read the temperature corresponding to D. The amount of superheat after the expansion is 300 deg. F., minus this temperature. Thus

$$300 - 187 = 113 \text{ deg.}$$

of superheat.

The area of the entropy-temperature diagram, like the indicator diagram, represents work; but on the former, the work is obtained in British thermal units and not in foot-pounds, as with the indicator diagram. As an example of area, the Rankine cycle may be considered. This is the ideally perfect cycle with which the cycles of ordinary engines are compared. The Rankine cycle is as follows: Water is heated from a temperature t_3 to a temperature t_1; it is then converted into steam at t_1 and expanded adiabatically to t_3. At the temperature t_3, it is condensed and returned as water to the boiler.

EXAMPLE 7. Show the Rankine cycle between temperature t_3 and t_1. What is the work done per pound of steam?

SOLUTION. In Fig. 8, opposite temperature t_3, mark D on the water line. Go up the water line to A, opposite temperature t_1. Proceed horizontally to the steam line at B. From B drop a vertical line to meet the horizontal line, drawn opposite t_3, at C. Then the Rankine cycle is given by

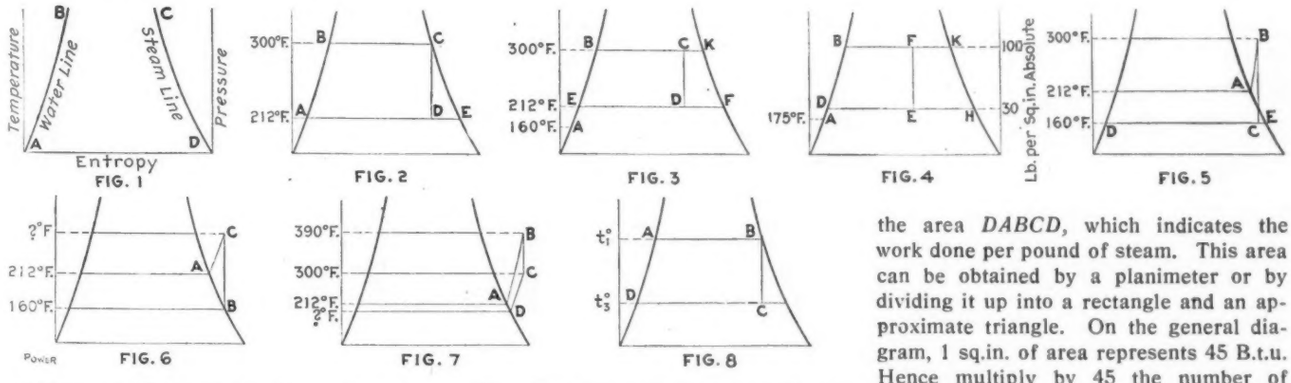


FIG. 1 FIG. 2 FIG. 3 FIG. 4 FIG. 5

FIG. 6 FIG. 7 FIG. 8

the area $DABCD$, which indicates the work done per pound of steam. This area can be obtained by a planimeter or by dividing it up into a rectangle and an approximate triangle. On the general diagram, 1 sq.in. of area represents 45 B.t.u. Hence multiply by 45 the number of square inches obtained and the result will give the work done on the Rankine cycle in British thermal units.

Enough examples have been given to show that no one of ordinary intelligence need have difficulty in working out various thermal relations on the chart. After becoming familiar with the simple chart given, it is an easy step to the later charts, which include in addition heat units and volume lines and use absolute temperatures. Instead of measuring with a rule to obtain the ratios or using a planimeter for the areas, the answer to any of the problems given may be read directly from the chart. It is, therefore, more convenient to use, although at first glance rather confusing.

In discussing the utilization of waste heat from gas engines, K. Kutzbach, in *Stahl und Eisen*, emphasizes that it is impracticable to attempt to allow the jacket water to exceed, say, 205 deg. F., owing to the danger of preignition, and that the exhaust gases have a large volume with comparatively low heat content besides corrosive impurities. Utilization may be wise, but it is limited.

Mixed Pressure Turbine Installation

By John S. Leese

When it became necessary to increase the capacity of the power plant of a large English flour mill it was decided to install a mixed-pressure turbine operating on exhaust steam from the reciprocating engines already in use.

The conditions which prompted this decision are described and reports given of the operating results, which exceeded the guaranteed performance.

The figures and data presented below are from a typical case in which the application of a low-pressure turbo-electric unit has successfully met the demand for increased power in an industrial establishment without involving the throwing out of good and expensive steam plant, or unduly encroaching upon the space at command.

The adaptation of the purely low-pressure turbine to work in conjunction with an existing steam plant where the additional power required is between 40 and 60 per cent. of the normal rating of the reciprocating plant, and the mixed-pressure type when the increased power required exceeds this amount, effects gains in economy which naturally vary with the conditions obtaining; but there are few applications in which the saving will not discount the initial expense in the course of three or four years.

The application at present under review is that of a large flour mill in the north of England, the size of which has gradually increased so that an addition to the existing plant was rendered necessary about three years ago.

The existing plant comprised five 8x30-ft., and two 8½x30-ft. two-flue Lancashire boilers fitted with automatic sprinkling stokers, forced draft, superheaters, and economizers. These boilers supplied steam at 120 lb. pressure and 150 deg. superheat to three reciprocating steam engines: "No. 1," a 22 and 38 by 36-in. vertical compound Corliss; "No. 3," a 19 and 35 by 60-in. tandem-compound Corliss; "No. 4," a 16 and 28 by 60-in. tandem-compound Corliss. These three engines, all running condensing, drove the mill by means of ropes.

When it became necessary to increase the plant capacity, conditions rendered it advisable to install electric drives for the new machinery. A generator was therefore necessary and for its drive the consulting engineers had two options.

One was to install a high-pressure steam engine and more boilers; the other to provide a mixed-pressure turbine to work in conjunction with the existing reciprocating equipment. The choice fell upon the latter and the following were calculated as the new duties and performances of the engines already installed:

Since there is no equivalent to an indicated horsepower in a turbine, for purposes of comparision, 1 i.hp. on the engines is taken as being equal to 1 b.hp. at the turbine shaft, although in reality 1 b.hp. equals approximately 1.08 i.hp.

With the above figures in hand, two proposals were considered. The first was to run as follows:

No. 1 engine condensing giving approximately	500 hp.
No. 3 engine non-condensing giving approximately	600 hp.
No. 4 engine non-condensing giving approximately	450 hp.
and the turbine condensing giving approximately	850 hp. (582 kw.)
and a total of	2400 hp.

On this basis, the turbine maker guaranteed an overall steam consumption of the plant not to exceed 12.4 lb. per indicated horsepower per hour.

The alternative scheme was to operate as follows:

No. 1 engine non-condensing giving approximately	500 hp.
No. 3 engine non-condensing giving approximately	600 hp.
No. 4 engine non-condensing giving approximately	450 hp.
and the turbine condensing giving approximately	1150 hp. (790 kw.)
and a total of	2700 hp.

A guarantee of 11.3 lb. per indicated horsepower per hour overall consumption was promised for this arrangement.

The first proposal, with the higher consumption guarantee, was chosen, since the turbine maker's guarantee did not cover liability in the case of failure of the reciprocating sets to run thus constantly heavily loaded and noncondensing.

A. Steam consumption when running on exhaust steam only, with an initial pressure at the turbine steam chest of 16 lb. absolute, unity dryness, and a 28-in. vacuum (30 in. barometer).

Load	890-kw. max. on L.P. end	Max. with 22,000 lb. 585 kw.	500 kw.	300 kw.
Consumption in lb. per kw.-hr.	32.1	37.7	40.6	51.7

B. Steam consumption with high-pressure steam only, with an initial pressure of 180 lb. per sq.in. absolute, superheated to 500 deg. F., and with a 28-in. vacuum.

Load	1650 kw. (150% normal)	1375 kw. (125% normal)	1100 kw. (normal)	825 kw. (75% normal)	550 kw. (50% normal)
Consumption lb. per kw.-hr.	16.3	15.8	15.3	16.4	17.7

C. Steam consumption when running as a mixed-pressure turbine on 22,000 lb. of exhaust steam per hour.

Load	1100 kw. (normal)	825 kw. (75% normal)
Live steam required to make up power, in lb. per hour.	6500	4100

Engine No.	Same cut-off Pressure 120 lb. Superheat 150 deg. I. hp. Condensing	Same cut-off Pressure 120 lb. Superheat 150 deg. I. hp. Noncondensing.	Total dry steam in L. P. diagram lb. per hour	Total steam = dry steam + 15% moisture lb. per hour	Consumption lb. per i.hp. per hour. Condensing	Consumption lb. per i.hp. per hour. Noncondensing	Exhaust steam available per i.hp. per hour
1	656	454	7,390	8,500	13.0	18.7	15.9
3	911	685	12,320	14,200	15.6	20.7	17.6
4	527	368	6,130	7,050	13.4	19.1	16.2
Total	2094	1507		29,750	average		
					14.2	19.7	

In placing the order for the mixed-pressure turbine, the following figures were specified:

The turbine was supplied with a multiple-jet condenser, the turbine itself being of the impulse type with a velocity wheel at its high-pressure end.

The accompanying table gives the results of recent tests upon the plant, the guarantee figures tabulated above being placed in an adjacent column for easy reference.

(This table is given on the next page.)

From these results it will be seen that the steam consumption when corrected for superheat is but 0.03 lb. per indicated horsepower per hour in excess of the lower guarantee contained in the second proposal.

Remarkable similarity is shown between the tested engine horsepowers, as

given in the first table, and the last-test results, the respective figures being:

For No. 1 engine condensing 656 and 653 hp.
For No. 3 engine non-condensing 685 and 684 hp.
For No. 4 engine non-condensing 368 and 372 hp.

Since the installation of the turbine the mill owners have been able to cut out one of the seven boilers. The increased capacity of the milling plant, the facility with which the grain can be handled in the processes, the extreme cleanliness of the factory, and the reduced line shafting and transmission losses are also advantages which have been made very apparent since installing the electrical equipment.

	Test	Guarantee
Average steam pressure, lb. per sq.in. gage	123	120
" " temperature, deg. F.	473	500
Back-pressure on engines, lb. per sq.in. absolute	17	17
Low-pressure steam at turbine steam chest	16.17	16
Engine No. 1.		
Indicated horsepower condensing	653	500
Steam consumption, lb. per hour	8,500	6,500
Speed, r.p.m.	79	79
Engine No. 3.		
Indicated horsepower non-condensing	684	600
Steam consumption, lb. per hour	11,500	12,400
Speed, r.p.m.	72.7	71
Engine No. 4.		
Indicated horsepower non-condensing	372	450
Steam consumption, lb. per hour	6,600	8,600
Speed, r.p.m.	70.5	70
Turbine.		
Brake horsepower	735	850
High-pressure steam consumption, lb. per hour	1,750	
Low-pressure steam consumption, allowing 15 per cent. for moisture, lb. per hour	15,400	17,850
Three Engines and Turbine.		
Total output, b.hp. and i.hp.	2,444	2,400
Total high-pressure steam consumption, lb. per hour	28,350	
Test Results.		
Overall steam consumption of plant, lb. per i.hp.-hour	11.59	12.4
Correction for 27-deg. superheat, 27 ÷ 12 = 2¼-per cent	0.26	
Corrected consumption, lb. per i.hp.-hour	11.33	
Difference between test and guaranteed consumptions, lb	1.07	
Per cent. improvement on guarantee	8.5	

The mixed-pressure turbine is governed by a positive-acting centrifugal governor which, through pressure, irrespective of speed, automatically controls the admission of live steam to the turbine, if the supply of exhaust steam should fall short of the load demand. The governor also prevents excessive speed variations when the load is suddenly thrown on or off the turbine, the momentary figure being 6 per cent. and the permanent figure 3½ per cent. of the normal speed when the load is suddenly applied.

An oil separator in the exhaust lines of the engines in front of the turbine inlet allows the condenser water to be used for boiler feeding.

The multiple-jet condenser has a rotary valveless dry-vacuum pump and a centrifugal extraction pump, both driven by a single 55-hp. motor mounted on the same bedplate. The condensing water used per hour at full load is 1900 gal., and its temperature is 65 deg. F.

The turbo-generator is of the pipe-ventilated type, rated at 1100 kw. at 2400 r.p.m., and generates three-phase current at 500 volts and 40 cycles. The shunt-wound exciter, which has a radial commutator, is directly connected and has an outboard bearing. The alternator ventilating air is cleaned by a cheesecloth filter. The generator is designed to stand overloads of 25 per cent. for two hours, and 50 per cent. for three minutes, with a maximum temperature rise of 60 deg. above that of the surrounding atmosphere. The following guaranteed efficiencies were exceeded in the electrical test:

Load	150% 1650-kw.	100% 1100-kw.	600-kw.	50% 550-kw.
Power factor 0.8	96	95½	93	93
" " 1.0	96½	96	94	94

The generator supply is controlled from a 12-panel black enameled slate switchboard, from which the power and lighting circuits are taken. The power circuits are at 500 volts pressure, and three-phase, 40-cycle motors are installed throughout the electrically driven part of the mill. Two step-down static transformers, of 100 kv.-a. capacity each, reduce the pressure to 110 volts for the lighting circuits.

The British Admiralty has ordered from the four principal Scottish mineral-oil companies 200,000 tons of oil fuel. The price is about 4½c. per gal., and most of the oil will be delivered at Rosyth base. Many destroyers and torpedo boats are being fitted for burning oil fuel exclusively.

The Government as a Coal Purchaser

Under the old plan of purchasing coal, George S. Pope explains, in Bulletin No. 51 of the Bureau of Mines, when the consumer found fault with its quality, he was assured that it must be good because it came from a mine with an established reputation. But as each dealer claimed that his coal was equal in quality to that of the others, there was no standard and the contract could not be cancelled. Purchased under the Bureau of Mines specifications, the consumer gets what he pays for. The advantages are:

1. Bidders are placed on a strictly competitive basis as to quality and price. This simplifies the selection of the most desirable bid and minimizes controversy in making awards.

2. The field for both the government and the dealers is broadened, as trade names are ignored and comparatively unknown coals offered by responsible bidders may be accepted without detriment to the purchaser.

3. The government is insured against poor and dirty coal, and from condemnation disputes based on visual inspection.

4. Experience with the old contract shows that it is not always expedient to reject poor coal, because of the delay and cost of removal. Under the present system, rejectable coal may be accepted at a greatly reduced price.

5. A definite basis for the cancellation of the contract is provided.

6. Constant inspection and analysis furnishes a check on the practical results obtained in burning the coal.

7. Paying on the quality basis incites the contractor to prepare the coal more carefully.

The government purchases annually from $6,500,000 to $8,000,000 worth of fuel. This sum includes the cost of delivery and of stowage. The Navy, War, Treasury, Interior and Commerce and Labor Departments are large consumers. Much of the government coal must be delivered by wagon, limiting the business to dealers having hauling facilities, particularly in Washington, where coal is purchased principally for heating the public buildings.

For the fiscal year 1908-9 the Geological Survey made tests and analyses representing 611,400 tons of coal bought under contract at an estimated cost of $1,858,800; for 1909-10, 829,300 tons costing $2,286,800; for 1910-11 the Bureau of Mines made analyses covering 1,091,400 tons, costing $3,084,800; and during 1911-12 it is estimated that the bureau will make analyses covering 1,500,000 tons, and costing $4,750,000.

"They call it an electric light," Pat confided to his companion, "but it do beat me how they make the hairpin burn in that little bottle."

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Sizes and Capacities of Wires

By G. H. McKelway

In the early days of the wire industry each manufacturer had his own gage but, in the interest of uniformity, the gages have been made standard, and the one at first used by the Brown & Sharpe Co. is now accepted for all sizes of copper wire up to and including No. 0000. When referring to this gage it is generally known as the B. & S. gage, from the initials of the firm bringing it out.

With this gage the higher the number the smaller the wire, and each size has 26 per cent. greater cross-sectional area than that of the wire with the next higher number. For rough calculation it is easier to remember that each size is approximately 1.25 times that of the next smaller. This rate of increase makes the wire three numbers lower, about twice the size of the first wire, so that, for instance, No. 7 has twice the section of No. 10.

Another point, easy to remember, is that a wire ten sizes larger than another has ten times its area and therefore one-tenth of its resistance, as the resistance of a wire changes inversely as its section.

For wire larger than No. 0000 it is customary to give the number of circular mils making up their area. A mil is 0.001 in. and a circular mil is the area of a circle 0.001 in. in diameter, so that a wire 1 in. in diameter has a section of 1,000,000 circular mils.

It is very unusual to draw solid wire larger than No. 0000, which is equivalent to 211,600 circ. mils, or nearly ½ in. in diameter, as it is then so stiff that it is very hard to handle. In fact, sizes of No. 0 and above when solid are seldom insulated and their greatest use is for trolley wire. For interior wiring, such as is done for lighting or power in buildings, it is not often that a solid wire larger than No. 8 or No. 10 is used.

In order to get the necessary flexibility with the larger sizes of wires they are each made up of a number of smaller wires stranded together. These wires are wound tightly together yet it is evident that they can never be so placed that there will not be small openings in their section and that all of the surface of each wire will be touching some other wire. Therefore, a stranded wire of the same diameter as a solid wire will not contain as much copper as the latter. For this reason the size of the stranded wire is calculated, not from the diameter of the wire as a whole, but from the sum of the areas of each of the strands. It follows that a stranded wire has always a larger diameter than a solid one of the same nominal size.

Having learned the ratio between the different sizes of wire it is only necessary to remember the resistance of one size and the resistance of any other can be easily calculated. Probably the wire whose size and resistance can be most easily remembered is No. 10, which is also a size much used in interior wiring. The reason for the ease in remembering the facts concerning it lies in the number of tens involved. Not only is the number of the wire 10, but it also has a diameter of 0.1 in., an area of approximately 10,000 circ. mils, and a resistance of one ohm in 1000 ft. of length. These figures are not exact but are close enough for all calculations as they only differ from the exact dimensions and resistance by 2 or 3 per cent.

Where large wires are concerned it is unnecessary to remember their area and diameter as the former is given by the description of the wire and the latter, in thousandths of an inch, is the square root of the area. To obtain the approximate resistance per thousand feet it is only necessary to divide 10,000 by the number of circular mils of the wire and the result will be the resistance in ohms. For instance,

$$\frac{10{,}000}{1{,}000{,}000} = 0.01$$

which is the resistance in ohms per thousand feet of a 1,000,000 circ. mil wire.

Another way of stating the same rule is to say that the number of feet per ohm is equal to the result obtained by dropping one cipher from the number of circular mils of the wire. By cutting off the last cipher from 1,000,000 circ. mils 100,000 is left, which is the number of feet of a 1,000,000 circ. mil wire needed to have a resistance of one ohm.

On comparatively short runs it is probable that the size of the wire will be limited not by the drop caused by its resistance but on account of the fire underwriters' requirements as to the amount of current that may be carried by the wire without its becoming too hot for safety.

The underwriters divide all insulated wires into two classes, depending upon whether or not the insulation used on them is of rubber, and in the rule books supplied by the National Board of Fire Underwriters is a table giving the amount of current allowed for each size of wire. As copper wire is almost universally used in buildings the tables are made out for wire of that metal, but the statement is made that aluminum wires of the same size and having the same insulation are permitted to carry only 84 per cent. of the current allowed for copper. This is because of the greater resistance of aluminum wire.

The amount of current which a No. 10 rubber insulated wire is permitted to carry is limited to 24 amperes while with any other type of insulation the capacity of the same wire would be 32 amperes, or one-third more. This ratio between rubber and other insulating material holds good approximately for all of the B. & S. sizes, although it increases a little as the sizes increase.

When the wires are 200,000 circ. mils or larger the ratio between the different types of insulation is as 1: 1.5, rubber being always the lower, not because wire protected with rubber becomes hotter than with other insulations or even because the temperatures a little higher than those allowed by the underwriters would be dangerous to new wire, but because continued high temperature deteriorates the insulation and rubber is more easily affected than the other materials.

The amount of current that a wire can carry does not increase directly with its area, so that a wire twice as large as another can carry only about 1.66 times as much current as the smaller one. Therefore, while the size of the wire doubles with every three numbers, the current should not double until the fourth number is reached, when it can safely do so.

The current carried by the wires of the B. & S. gage and not smaller than No. 16 can be calculated from the amount permitted to be used on No. 10, as previously given.

When working with wires whose area is great enough to be given in circular mils, it should be remembered that a 200,000-circ. mil rubber-covered wire is permitted to carry 200 amperes, and that with other insulation 300 amperes is allowed. The capacities of other sizes can be readily found by allowing two-thirds more current for every time that the cross-section of the wire is doubled.

The Small Portable Storage Battery

By B. Dawson

Although battery manufacturers are constantly sending out instructive circulars regarding the care, charging rate, discharge rate, capacities, etc., of their respective products, yet the average user of the small battery keeps violating the instructions given.

A common mistake is to use the battery until the last bit of spark has gone out of it. Right here is where more or less damage is done to the plates by excessive sulphation. Unless proper precautions are used to rid the plates of the sulphate, the battery will more and more rapidly deteriorate.

In choosing a good portable storage battery, preference should be given to one having each cell terminal exposed so that it is easy to test the voltage of each cell during the charging period. It is also essential that the terminals be in good condition, free from the effects of the containing acid or charging acid gas in the form of creeping salts. Good-sized vent holes should be on the battery to allow easy inspection of the plates and the electrolyte in each cell; this will also allow the gases to escape freely during the charging.

The strap handle should be strong and durable and of the detachable kind. The strap handle should always be removed, except for carrying the battery from place to place.

Rubber jars in a battery have the advantage over glass jars in that the former will stand more shock; otherwise the glass jars are just as good. The positive and negative plates should amply comply with the manufacturer's rating; therefore it is essential to choose good reliable makes that will not disintegrate too easily on maximum discharge. Herein lies the difference between a cheaply constructed pasted plate, and a first-class one.

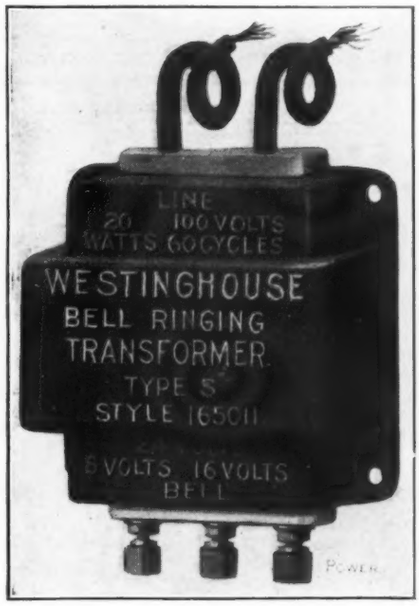
The writer has found the following to be common faults in the operation of the small portable storage battery: Dirty battery terminals; stranded and dirty wires connecting the leads; insufficient charge in battery; low capacity caused by sulphated plates, or low water in the cells; one or more short-circuited cells caused by sediment touching the plates, thereby lowering the terminal voltage; too small a battery for the required service; tampering with the electrolyte, such as spilling it or adding contaminated water, and one or more cracked jars, causing the electrolyte to slowly dribble away and ruin the outer casing.

Following are a few recommendations for the care of small portable batteries: Keep the batteries clean by washing all around the casing with pure water, taking care to keep water out of the cells. Keep the battery and terminals perfectly clean and make good connections. Never allow the battery to discharge below 1.65 volts per cell. Always recharge the battery immediately when the minimum (1.65 volts) has been reached. Never put the battery away for the winter in a discharged or low-water condition. When the battery is put away for the winter make sure it is placed in a dry, nonfreezing room, and is fully charged with sufficient distilled water added to cover the plates. Keep the vent plugs well sealed during this time to prevent undue evaporation; this will also keep out the dirt. If the batteries are needed in the spring a freshening charge should be given.

Do not let inexperienced or careless workmen charge or repair a battery. Avoid taking the batteries to repair shops where machine-shop bench-work and battery material are all handled from the same bench. Use only pure distilled water to replace loss by evaporation.

Bell Ringing Transformer

To supply electric bells from a lighting circuit, the Westinghouse Electric & Manufacturing Co. has placed on the market a new transformer, built according to the design of its well known type S distributing transformer. It is light, compact and absolutely fireproof, and will deliver on open circuit 8, 16 or 24 volts, or 16 volts fully loaded. It is particularly useful for apartment houses and public buildings.



Westinghouse Bell Ringing Transformer

The coils and magnetic circuit are vacuum dried and impregnated with a moisture-proof insulating compound. After this treatment the complete unit is fitted into its case. The interior of the case is then filled with a specially treated cement completely embedding the coils and iron in a fireproof and indestructible compound, which precludes any possibility of the bell-ringing circuit coming in contact with the 110-volt circuit.

Rubber-covered primary leads are brought out of the top of the case through a porcelain bushing and are left of sufficient length to permit connection to the lighting circuit. Binding posts are provided at the bottom as shown for the bell circuit.

CORRESPONDENCE

Station Inspection

The manager of the 2300-volt, three-phase, lighting system with which I am connected has inaugurated a system of tests and inspections which I believe to be indispensable to the upkeep of an electrical system.

Every three months the transformers, fuse boxes and lightning arresters are inspected. The height of oil in the transformers is noted and more added if necessary. The oil is tested for moisture by inserting a red-hot nail, moisture being indicated by a sizzling noise. Fuse boxes are inspected for loose contacts. When our system was first installed one set of secondary fuses blew quite often, and an inspection showed loose contacts which when tightened eliminated the trouble. The knurled cylinders of the high-tension lightning arresters had several burned spots, showing that they had been in action.

We test the machines and outside lines for insulation resistance with a megger. Starting at the power house we test the armature and fields of the exciter and alternator. With the machines exactly alike, one showed five megohms lower resistance than the other. The outside system, comprising four transformers and about 12,000 ft. of line, showed 300,000 ohms.

We next test each transformer and the lines radiating from it, removing the primary and secondary fuses. The transformer reading was infinity, the secondary lines testing from 40,000 to 100,000 ohms. The secondary lines included the interior wiring and fixtures.

The value of these tests is shown by the fact that one of our consumers claimed to have a ground on his system which caused his meter to record more kilowatt-hours than were used. Our test of his wiring showed 1½ megohms resistance, thereby disproving his contention.

The greatest value, of course, of systematic tests is the knowledge of defects, which in many cases can be remedied, and of impending breakdown.

Cecil R. Coile.

Fort Morgan, Ala.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

German Progress in Large Oil Engines

By F. E. Junge

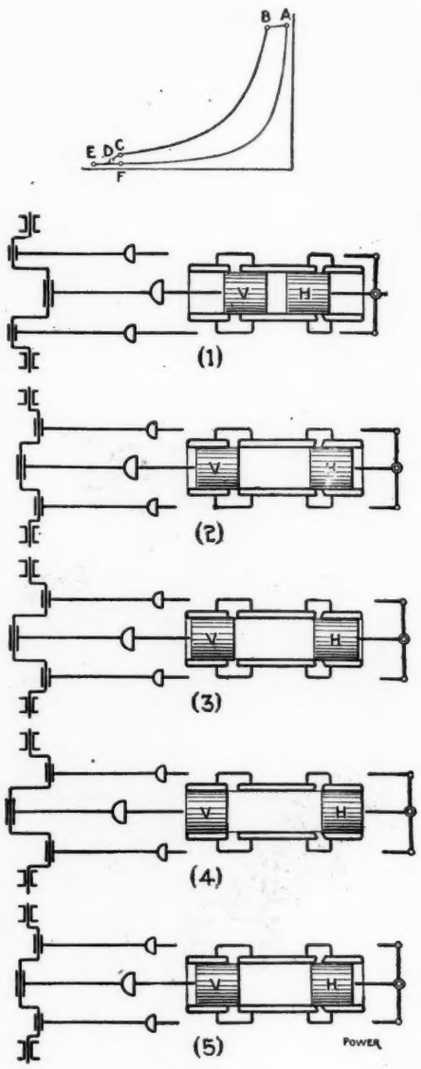
It is the purpose of this series of articles to bring American readers in close touch with the latest achievements of oil-power engineering in Europe, and especially in Germany, where, owing to the natural adversity of fuel conditions, and to the need of economizing in operating and other costs, power production has been developed to a high degree.

There seems to be a widespread impression in this country that the evolution of oil power in Germany has been and is associated exclusively with the development of the Diesel engine. While the Diesel principle has evolved a new era of power engineering, it would be unfair to slight the many individuals and firms who have taken an active part in the later development and have improved upon the original idea.

Of late much attention has been concentrated on the Junkers oil engine, claimed by the inventor, Professor Junkers, to be an improvement on the standard type of Diesel engine, an improvement in mechanical equipment, thermal efficiency, and capacity for overload. This engine operates on the Diesel cycle—using the term in the colloquial, not in the scientific sense—and was described by the writer in the Jan. 23, 1912, issue of Power. To make the present series complete, however, and for the benefit of those who have not read the former article, the general features of the Junkers engine will be reiterated, to be followed by details and a discussion of the design, leaving it to Professor Junkers to propound and to defend his scientific convictions.

The Junkers Engine

The Junkers engine works on the two-stroke-cycle principle, its method of operation being represented by Figs. 1 to 5. The cylinder is open at both ends and contains two pistons which are linked with a three-throw crankshaft, permitting them to move in opposite directions. In Fig. 1 the pistons are shown at their inner dead center, the space between them being filled with highly heated air, as a result of the previous compression stroke. In this position the fuel (oil) is injected into the compressed air in a fine spray, igniting and burning under constant pressure during the first part of the outward stroke from *A* to *B*, of the working diagram. From *B* to *C* the expansion of the burnt gases takes place, and at *C* the pistons have reached the position indicated by Fig. 2, in which the front piston *V* is just beginning to open a series of slots or ports, permitting the burnt gases to exhaust into the atmosphere. When the position shown in Fig. 3 is reached



Figs. 1 to 5

the equalization of the internal pressure with that of the atmosphere is almost completed. Simultaneously the rear piston *H* uncovers its series of slots, permitting fresh air at low pressure to enter the cylinder and expel the remaining burnt gases through the exhaust slots. This scavenging is continued until the pistons have passed the outer dead center (Fig. 4) and taken up the position shown in Fig. 5. At this instant both rows of slots are closed by the pistons traveling toward their inner dead center, a distance which is represented by *EF* in the diagram; at point *F* the cylinder is filled with fresh air, cut off from the atmosphere. As the pistons approach one another in traveling toward the inner dead center (*FA* in the diagram), the contents of the cylinder is compressed and heated to such a degree that the fuel injected shortly before point *A* is reached, ignites and the cycle is repeated.

For large work the tandem arrangement is used, which reduces the weight, cost and space of the engine per unit. In this arrangement a three-throw crank is used, the two outer pistons being linked with the central crank, and the two inner ones to the side cranks, which are placed at 180 deg. to the former. This is represented in plan and elevation by Fig. 6. By this arrangement each stroke is a working stroke. While the pair of pistons in one cylinder are executing an outward movement (power stroke), the pair in the other cylinder are traveling toward each other (compression stroke). When the pistons of one cylinder occupy the inner dead center, those of the other cylinder take up the outer dead-center position. In this way, while the fuel injection is taking place in one cylinder, scavenging is going on in the other. Scavenging is accomplished in the same way as in the single-cylinder engine.

The governing of the slots is performed by the pistons, complicated governing elements being thus dispensed with. Arranged symmetrically to the center line of the cylinder are the two scavenging pumps, which supply the air for scavenging, and the four-stage air compressor for the fuel spray air is similarly arranged; these auxiliaries are actuated from the yoke between the middle pair of pistons. The details of construction will be given later. For large work each cylinder has two fuel valves and one starting valve for compressed air. The fuel injection is performed in the ordinary way by means of highly compressed air, the atomized fuel being brought into close contact with the air provided for combustion.

As can be inferred from the schematic representations of the Junkers engine, the cylinders are simple castings, compared with the more complicated cylinders of the standard Diesel engine. The acting

forces balance one another in the power-transmitting mechanism, whereby an almost complete balance of the inertia of the reciprocating parts is attained. The bedplate need only extend up to the first cylinder, the cylinders themselves being carried on flexible supports and resulting in a comparatively light and inexpensive engine. The cylinders have no covers and no stuffing-boxes; hence on one side the pistons are always in contact with the atmosphere. Moreover, with every outward stroke they enter into well cooled parts of the cylinder, which are not touched by the products of combustion. This mode of operation secures a perfect lubrication of the pistons and cylinders, the oil being admitted at points, not in contact with the burnt gases.

Owing to the division of the total stroke between two pistons a high aggregate piston speed is attained, as well as a high rotative speed, and comparatively low separate piston speeds. Another feature is the long cylinder of relatively small diameter, offering exceptional scavenging opportunities and providing a favorable combustion chamber. A scientific discussion of these various features will be given in following articles.

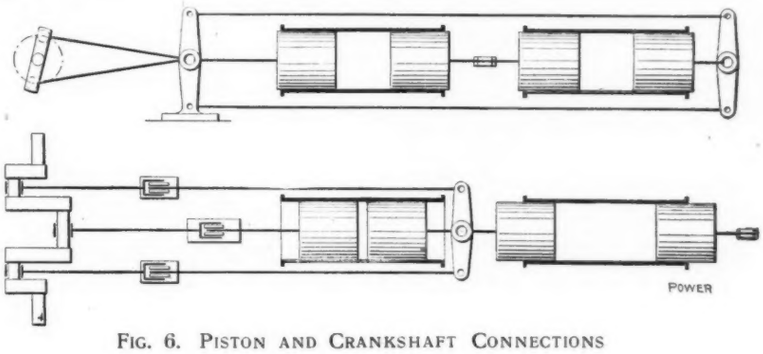


FIG. 6. PISTON AND CRANKSHAFT CONNECTIONS

Two-stroke Cycle Scavenging

BY ROBERT MILLER

All modern two-cycle engines operate on the Clerk cycle, or slight modifications of it, which are mostly mechanical.

The Clerk engine had a separate pump for handling the new charge which entered at the head end of the power cylinder. One modification uses the crank end of the power piston and incloses the crank space, so that on the outstroke this piston has combustion on the cylinder side of the piston and a slight compression of the fresh charge on the crank side. On the instroke, it has compression of the new charge on the cylinder side of the piston, and the suction of a fresh charge in the crank chamber. Other modifications use separate pumps for handling the air and the gas. Sometimes the new charge is delivered to the power cylinder at the head end; sometimes through ports uncovered by the piston head at the end of its outstroke.

The scavenging charge enters the power cylinder either as a blast or as a slow displacement. Where the power piston acts as both a power and a pump piston it is supposed to give the blast effect; where the pump pistons are timed half a stroke in advance of the power piston the other effect is desired. In all cases the expulsion of the residual gases is believed to be by the displacement due to the entering charge. And it is because this displacement is imperfect at its best, and unsatisfactory under varying conditions of speed, load and throttling, that the two-stroke-cycle has not, after 30 years, displaced the four-stroke-cycle or even held its own, except for certain classes of service.

When the piston of a four-stroke-cycle engine travels inward on its exhaust stroke, it pushes ahead of it a column of exhaust gas. At all speeds, loads and throttle openings, this piston travel is the same, so that the scavenging is positive, and always the same, neglecting the slight variation due to wire-drawing.

The simplest form of two-stroke-cycle is the crank-case compression with piston uncovered ports. Near the end of the explosion stroke, the piston uncovers the exhaust port; the excess pressure in the cylinder is relieved at this point until the further movement of the piston uncovers the fresh-mixture inlet port. Depending upon the size of the exhaust port, its lead over the inlet port, the pressure in the cylinder, and the speed of the engine, three conditions may exist. The pressure in the cylinder may be greater, in which case part of the cylinder contents will pass into the crank case until the two pressures are equalized. Moreover, the temperature of the cylinder contents may be high enough to ignite the fresh charge. While this equilibrium is being established, the exhaust port is still open, so that while the crank-chamber contents flow from the admission port into the cylinder, there is a flow from the cylinder through the exhaust port. It is usual to place a deflector on the piston head to direct the crank-chamber contents to the cylinder head; but with the pressures in the cylinder and crank chamber equal, there is no cause for the new charge to travel to the head when it is much easier to pass over the deflection to the open exhaust port.

When the pressure in the cylinder and the crank case are equal at the moment of inlet-port opening, the same action occurs as in the first case.

The third condition is entirely different, yet it is what is usually assumed to be true. The cylinder pressure being lower than the crank-case pressure the new charge passes out of the bypass port and strikes the deflector with considerable velocity. A small portion gets over the top and goes directly to the exhaust port; the major portion is directed upward. There is a volume of burned gas in the cylinder either quiescent, or moving slowly toward the exhaust port. The new charge moving rapidly meets this mass; so that the action instead of being a real displacement is a quick intermingling. In fact, this blast may pierce the old charge and displace very little of it.

Where the charge enters at the head end it is usual to provide valves which are opened by the greater pressure of the new charge, or mechanically by cams or eccentrics. A few employ piston valves.

Where the valves are in the head, the new charge does not directly displace the volume directly behind the valve (poppet type) and the stream is in the form of a hollow cylinder. The clearance-space regions remote from this stream are effected by it only by an eddy action and have no real displacement. In some cases the new charge has been given a swirling action, in an attempt to reach the distant regions of burnt gas.

Where a blast action is not desired, but rather a gradual displacement, the action is somewhat as follows: At the moment the inlet opens, the cylinder and displacing pressures equalize; then, depending upon the shape of the valves or other impediments in its path, the charge passes in straight lines toward the exhaust ports, the surrounding residual gas gradually passing into the column.

All of these actions will vary because of the four reasons stated. The size of the exhaust port and its lead over the inlet port, determine the cylinder pressure at the moment of inlet opening. The pressure in the cylinder being dependent upon the throttle, open throttle means high exhaust pressure. The speed affects the time available for the exhaust to pass out, as double the speed allows only one-half the time.

Thus far displacement by full charges only have been considered. When the charge is throttled, more complications result, as the pressure balance is again disturbed. In addition, small new charges displace small quantities of old gas, leaving more old gas to foul the new. In a four-stroke-cycle, with the smallest throttle opening at which the engine will run, the exhaust displacement is the same as at full throttle, so that the small fresh charge is only slightly fouled by the burnt gas in the clearance.

The Blanchard Oil Engine

Among the latest additions to American oil-engine practice is the Blanchard engine built by the Blanchard Machine Co., of Cambridge, Mass., in sizes of from 8 to 100 hp. This is a slow-speed heavy-duty engine operating on the two-stroke cycle with kerosene, distillate or fuel oils. Fig. 1 shows the general appearance of a single-cylinder Blanchard engine, although they are made with as many as four cylinders.

Only moderate compression is employed and as the oil is sprayed into the combustion chamber, is vaporized by the hot air and ignited by a hot tube, relatively slow combustion takes place; hence the engine is not subjected to such extreme pressure and shocks as prevails in the ordinary gas or gasoline engine.

Considering the cycle of operation of the Blanchard engine, Fig. 2 represents a vertical section through the cylinder with the piston at the bottom of its stroke. As it starts on the upstroke it covers the inlet port *A*. The crank case is air-tight and as the air-intake port *B* is always covered except when the piston is at the top of the stroke, a partial vacuum is formed within the crank case. When the piston reaches the top of the stroke and uncovers the air-intake port, air is sucked into the crank case through the pipe *C*. On the down-stroke this air is compressed to about 5 lb. Just after the exhaust port *F* has been uncovered the piston uncovers the transfer port *D* and the inlet port *A*, permitting the air confined in the crank case to rush through the passage *E* and, deflected by the piston, into the upper part of the cylinder. This serves to scavenge the cylinder. Upon the next up-stroke this fresh air is compressed to about 200 lb. (the exhaust and inlet ports being covered by the piston).

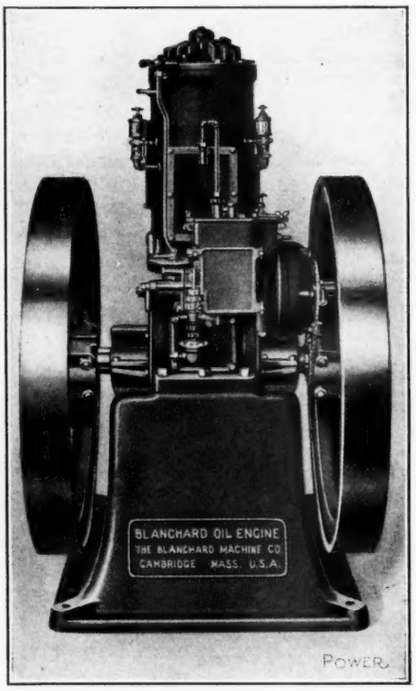


FIG. 1. SINGLE-CYLINDER BLANCHARD ENGINE

The heat of compression, together with that absorbed from the cylinder head, which is unjacketed, is sufficient to raise the temperature of the air to such a degree that the oil sprayed in is instantly gasified and is ignited by the hot plug *G*. In starting the engine this plug is first heated by the blow-torch *H*, but after the engine is in operation this is kept hot by the burning of each successive charge.

The injection of water into the combustion chamber forms a special feature of this engine.

Fig. 3 is a section through the fuel feed mechanism, the oil pump on the left and the water pump on the right, *I* and *J* being the oil and water plungers respectively. Both are driven by the camshaft *K*. By pushing in or pulling out the piece *L*, which is held by a set-screw, the stroke of the water plunger is varied. In this way any desired proportion of oil and water can be had. This proportion then remains fixed, although the quantity of fuel admitted is determined by the governor, which regulates the length of stroke of the pump plungers. The lever *M*, Fig. 2, is for priming.

The governor is of special design and is guaranteed to regulate within 3 per cent. from no load to full load.

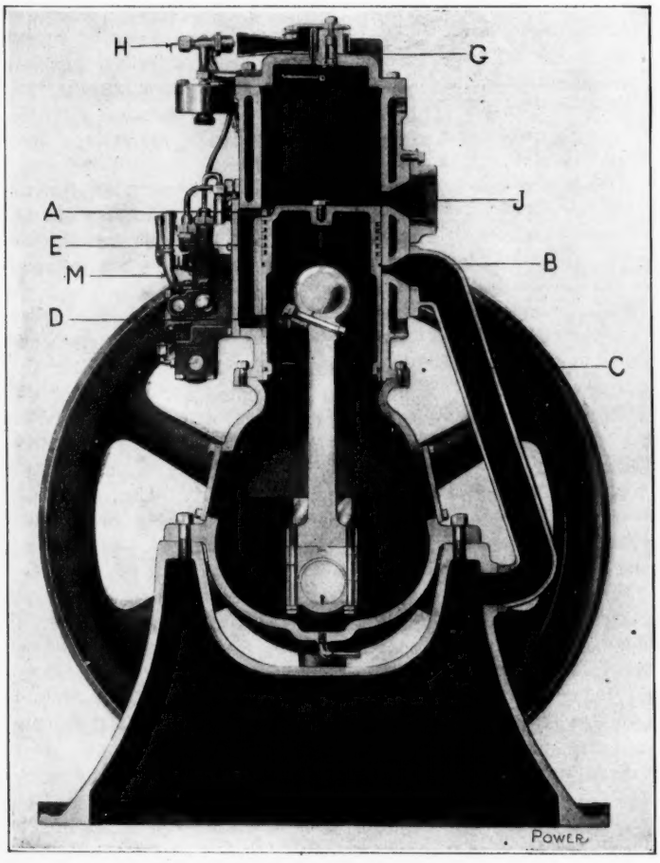


FIG. 2. SECTION THROUGH CYLINDER

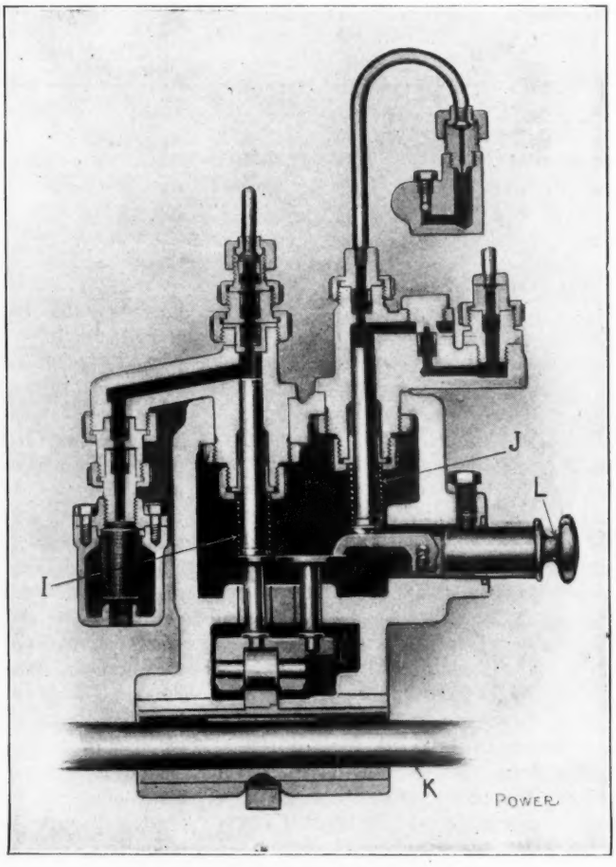


FIG. 3. FUEL AND WATER PUMPS

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Indicating the Ammonia Compressor

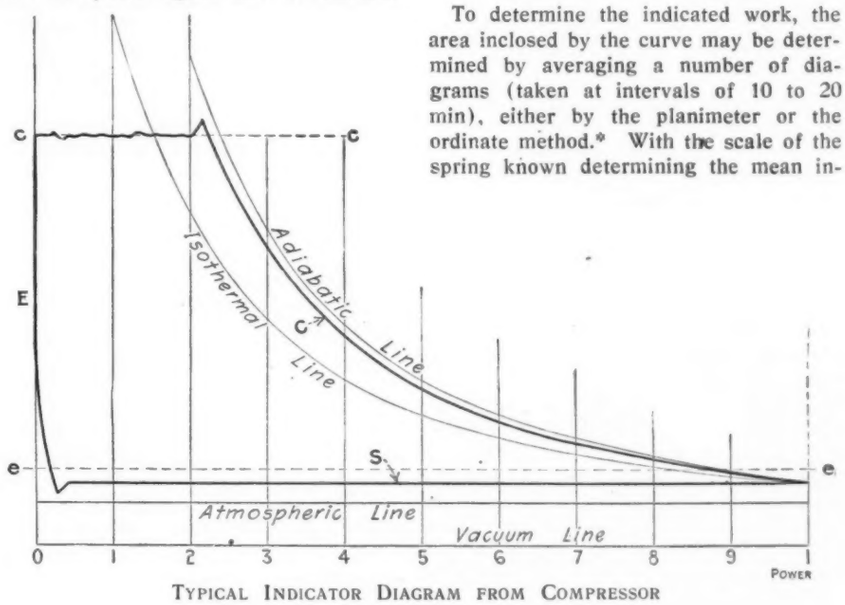
By John Creen

Regularly indicating a compressor and knowing how to correctly interpret the diagrams obtained is of great value. The indicator diagram is an important permanent record of the quality and quantity of work done by the compressor relative to its capacity. Diagrams should be taken every two weeks or so, and filed for future reference. For ammonia machines a special indicator entirely of iron and steel, fitted like the ordinary steam-engine indicator, is generally advisable. The ordinary bronze or composition indicator can be used without fear of damage, if the piston is removed, thoroughly cleaned, and the indicator cylinder kept well oiled with clean compressor oil after finishing the test. Diaphragm indicators, owing to the irregular bending of the diaphragm, have not given good results compared with piston instruments, although they let no ammonia escape.

The best compressor manufacturers provide their cylinders with indicator pieces bored along their whole length to the interior of the cylinder and generally sealed by a setscrew and flange besides having a thread for the indicator cock. Regrettably, up to the present, no definite standard for these threads and connections has been adopted. Very often intermediate pieces have to be procured to suit the threads before the indicator can be applied. This presents difficulties where, in order not to increase the clearance, the indicator opening is closed by a long pin not easy to remove. It would be better if the openings had a flange carrying both the thread receiving and indicator cock and the bored pin for filling the clearance.

With carbon-dioxide machines the pressure becomes so great that the ordinary indicator piston of ¾ in. diameter cannot be used. A smaller piston of 0.25 or 0.4 in. diameter, working in a lower narrow part of the indicator or in a box screwed therein, must be employed. These are commonly called Riedler pistons. The high pressure of the carbon dioxide also makes applying the normal indicator cocks very difficult; using small screw-down valves is better. These need not have a bore greater than ⅛ in., but should have a blow-through plug fitted to the cavity in connection with the indicator to restore atmospheric pressure under the indicator piston after taking a diagram.

Before the indicator cock is fixed to the compressor, the suction valve is closed and the machine allowed to make a few revolutions, so that most of the cylinder content is exhausted. The machine may then be stopped, the discharge valve closed and the indicator cocks or valves, as the case may be, screwed in position. The air which has entered the cylinder must finally be expelled by opening the cocks, and to a slight extent the suction valve, after which the discharge valve is opened again and the machine started. As soon as the suction valve is full open, a reading may be taken.



Typical Indicator Diagram from Compressor

The Indicator Diagram

On the diagram obtained the straight lines corresponding with the condenser and evaporator pressures must be drawn as soon as possible, for which purpose the manometer must be previously compared with the indicator. If all parts of the machine are in order, a diagram somewhat like the accompanying illustration will be obtained, the suction line *S* lying a little below the evaporator pressure *ee*, and the pressure line somewhat above the condenser pressure *cc*. The work required for opening the two valves is shown by the two little peaks, one on the suction line pointing downward and the other on the pressure line pointing upward.

The influence of clearance is seen from the back expansion curve *E*, which cuts the evaporator line *ee*, as shown, and diminishes the actual suction volume, quite apart from the fact that the vapors are not at the evaporator pressure, but at the somewhat lower suction pressure. The compression curve *C* in the diagram reaches the evaporator pressure only after the change of stroke, so that instead of the theoretical volume indicated by the length of *ee* the compressor has a smaller suction volume for vapors at the evaporator pressure.

Horsepower of Compressor

To determine the indicated work, the area inclosed by the curve may be determined by averaging a number of diagrams (taken at intervals of 10 to 20 min), either by the planimeter or the ordinate method.* With the scale of the spring known determining the mean indicated pressure is easy, and multiplying this by the area of the piston in square inches, gives the mean piston pressure in pounds. As the piston rod reduces the piston area on one side, this process must be carried out separately for each side. If then the mean value of the two piston pressures is multiplied by the stroke in feet and twice the number of revolutions per minute, the work in foot-pounds is obtained, and this divided by 33,000 gives the horsepower.

For example, take an ammonia compressor with a 10-in. diameter cylinder, a 2-in. piston rod and a stroke of 15 in. = 1.25 ft. The effective piston area is 75.4 sq.in. on the piston-rod side and 78.54 sq.in. on the other. If the mean indicated pressure is 41.54 and 42.80 lb. per sq.in. respectively, the total piston

*See Power, June 18, 1912, page 881, "Doing Without a Planimeter."

pressures will be 3132 lb. and 3361, the mean being 3246 lb. Then

$$\frac{2 \times 60 \times 1.25 \times 3246}{33{,}000} = 14.75 \; hp.$$

Defects Indicated by Diagram

The indicator diagram is useful in pointing out faults in a compressor. Too large a clearance flattens the back expansion curve, and means that the suction volume is being given insufficient use. A similar effect is produced by the jamming open of the discharge valve, caused by the spindle taking an oblique position in its guide. This defect frequently causes a late opening of the discharge valve and naturally a large expenditure of work, which is shown on the diagram by a large peak in the pressure curve. After opening, a loose valve immediately fixes itself wide open, and only when it shuts at the instant of the change of stroke does the defect come to light. With a long stroke the pressure valve may rebound from its seat; this is shown by a hook in the back-expansion curve, which runs very level from this point. Insufficient use of the stroke volume is the consequence. If the back-expansion curve proceeds normally after the hook, nothing is wrong with the play of the valves. The indicator piston clutching or its spring vibrating may have been responsible. Losses also arise from too high resistance in the suction and discharge pipes, too strong tension on the valve springs, faulty adjustment of the valves or the presence of an air cushion. These defects are indicated by too wide a variation from the evaporator or condenser pressures. In such cases the valve springs must first be replaced by weaker ones or more play be given to the air buffer; then if no appreciable improvement is obtained, constrictions should be looked for in the pipes and removed if any are found.

Jamming of the suction valve in the closed position will cause a considerable fall in pressure before the commencement of suction and the formation of a large peak on the suction curve. If the valve jams in the open position, compression is greatly delayed; that is, part of the stroke of the piston is wasted.

Much looseness of the valves is shown by the disappearance of the horns or peaks. The compression curve is steep and almost rectilinear when the pressure valve is the looser, and flat when more looseness exists in the suction valve.

When the piston does not fit tightly, the gas passes from one side to the other so that the pressure rises too slowly on the compression side and the compression curve becomes very flat. The late, and therefore sudden, opening of the discharge valve in this case is immediately recognizable by the vigorous banging of the valve on its seat. On the other hand, the pressure on the suction side, when the piston changes its direction, easily remains higher than that in the evaporator, and so prevents the suction valve opening. This lasts until the piston velocity has become so great that the leakage past the piston is no longer able to fill the suction space. The pressure then gradually falls and the suction valve begins to work. If several of the above defects are present, very involved diagrams may be produced; but their interpretation should not be difficult to the experienced engineer. Regular waviness of the various curves is generally due to indicator spring vibrations and not faulty valve action.

Most engineers are familiar with indicator diagrams and the meaning of their various lines; the admission line parallel to the atmospheric line, the compression line rising in an easy curve to the discharge line, and the latter usually wavy or peaked, due to vibration of the discharge valve. The admission and discharge lines should be joined by a nearly vertical line and the diagram should have a square heel, this representing the amount of clearance. But what the engineer needs is a guide, something he knows is correct with which to compare his diagram. The only true guide for comparison is the adiabatic line.

To plot the adiabatic line on the diagram one must have a table of adiabatic constants. The diagram is divided into ten equal parts. From the table of constants in the horizontal line is found the number corresponding to the absolute back pressure on the diagram; then, in the same vertical column which contains the absolute back pressure, are the values of p_1 to p_9. These quantities are laid off on the lines drawn on the diagram to the scale of the indicator spring; this gives a series of points through which a smooth adiabatic curve is drawn.

If the ammonia gas could be compressed from the beginning of the stroke to the condenser pressure in a perfectly tight and nonconducting cylinder, without loss or gain of heat, the indicator pencil would trace an adiabatic line. With no leakage past the piston or valves, the adiabatic line for all ammonia compressors should almost overlie the compression line of the card for its whole length. The water jacket seems not to affect the compression line materially, but, during the last quarter of the stroke when the gas is very hot, the adiabatic line may be slightly above the compression line. If the latter is not closely adiabatic, something is known to be wrong with the piston, valves or gaskets of the compressor, if the indicator has been properly connected, the pipe leading from the cylinder to the indicator is short and of small bore, say ⅛ in. diameter, and this pipe is well insulated from the cooling effect of the jacket water.

If an engineer periodically indicates his compressor as well as the steam end, and makes himself familiar with a simple and practical method of plotting the adiabatic line on the diagram, he will be able to get much better work from his machine.

CORRESPONDENCE

Drawing Charge from Double Pipe Condensers

As most refrigerating engineers know it is difficult to withdraw the charge of ammonia from a system using double-pipe condensers. The ammonia seems to lay up in the condenser and will not come down to the receiver.

In a 100-ton ice plant we had a 12-stand condenser 12 pipes high. Two pipes in each stand were 2½ in. in diameter with a 1¼-in. water pipe inside; the other 10 pipes were 2 in. in diameter with a 1¼-in. water pipe. The condenser was about 40 ft. above the liquid receiver which was located on the ice tanks close to the expansion valves.

On the first pump down I drew out all the liquid in the receiver, which amounted to about 35 lb. I knew that there was a great deal more liquid somewhere as there should be about 2000 lb. Upon investigation the liquid was found in the condenser coils.

To get the liquid down to the receiver, I closed the valve at the receiver on the liquid line feeding the expansion valves and then opened the expansion valves wide. I closed the gas and liquid lines on four of the 12 stands of pipe in the condenser and opened the auxiliary or pump outline to these four stands. The auxiliary line was connected to the suction side of the compressor, and I pumped down the system and these four stands of pipe to atmospheric pressure, discharging all the gas into the other eight stands of the condenser. Shutting down the compressor I let the gas condense and after a while drew off all the air and foul gas in the condenser through the purge valves, and the liquid into the receiver. Then I cut out four more stands of pipe on the condenser and proceeded in the same manner as before until the last pump down when all the stands of pipe on the condenser were cut out except one, which was used to condense the gas.

The system and the eleven stands of pipe were pumped down to 15 in. of vacuum and the remaining liquid was drawn out into the receiver.

C. E. Anderson.

Chicago, Ill.

Correction—In Mr. Anderson's article on "Setting the Expansion Valve," issue of Sept. 24, page 458, eighth line, an odd misprint occurred. The line should read: close the expansion valve and *pump down the coils*, etc.

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POWER

The Value of the Horsepower

When the steam engine began to assume industrial importance, it became necessary to have some unit by which to designate its capacity. The work to which it was initially put had been for the most part done by horses, and the capacity of the engines was naturally compared with that of the horses which they replaced. Savery in 1702 considered that an engine which would raise as much water as two horses working continuously would replace ten horses, and hence was a ten-horsepower engine, since relays of horses must be used to keep the work going continuously.

When Watt and Boulton got into the engine business they established a standard of their own. Some heavy horses from Barclay & Perkins brewery, London, were caused to raise a weight from the bottom of a deep well by pulling on a rope passing over a pulley. It was found that a horse could raise a weight of 100 pounds while walking at the rate of 2.5 miles per hour. This is 22,000 foot-pounds per minute. Watt added fifty per cent. for friction and good measure and established the value of 33,000 foot-pounds per minute or 550 foot-pounds per second.

But if Watt's well had been in New Orleans, and if his methods of measurement had been sufficiently accurate he would have found that with the same effort his horses could have lifted more than they could at London. The attraction of the earth upon the same mass is greater at the poles than at the equator, greater at the sea level than at an elevation.

This is due to the fact that on account of the greater diameter of the earth at the equator, a body there is further from the earth's center, as it is also at an elevation, and to the fact that at the equator the effect of centrifugal force which opposes that of gravity is greatest, becoming nothing at the poles.

Watt's definition of a horsepower is therefore not a fixed definite unit, but has a different value for every latitude and elevation. If we take it at 550 foot-pounds per second at 50 degrees of latitude (about that of London) it will become 549 foot-pounds per second at the poles and 552 at the equator. At 45 degrees it would be 550.48 at sea level, 550.45 at an elevation of 5000 feet and 550.61 at an elevation of 10,000 feet.

While these small differences are not likely to be serious in ordinary commercial transactions, it is not right nor scientific that so important a unit as that of power should have a changeable and unstable value.

A watt is the power developed by the action, with the velocity of one meter per second, of a force capable of giving to a mass of one kilogram in one second a velocity of one meter per second. Since the kilogram is a definite mass (that of a cubic decimeter, or liter, of pure water at the temperature of its maximum density) the force which will accelerate it at a given rate is also a fixed quantity, and the rate at which work is done when the force is exerted at a velocity of one meter per second is a definite, absolute unit, independent of latitude, altitude or any of the disturbing factors which complicate the unit determined by the effect of gravity.

The watt, or kilowatt is thus as much a mechanical as an electrical unit and may be used to indicate rates of power development and application even when no question of electricity is involved with more accuracy and as much propriety as the horsepower. Its use for this purpose should be encouraged, but since most mechanical determinations in English-speaking countries are made in foot-pounds, it is necessary that there should be a definite relation established between the gravitational unit, or horsepower, and the absolute unit or kilowatt.

The round number usually employed for this equivalent is 746 watts, which happens to equal 33,000 foot-pounds per minute for the latitude of 50 degrees, nearly that of London where Watt's experiments were made. By defining the horsepower, therefore, as 33,000 foot-

pounds per minute at 50 degrees latitude and sea level, its equivalence with the absolute system of units is established as 746 watts.

The United States Bureau of Standards has adopted this value and, in a recently issued circular entitled the "Relation of the Horsepower to the Kilowatt," discusses the question historically and otherwise giving tables of the value corrected for different latitudes and elevations and urging the use of 746 watts as the exact equivalent of the horsepower instead of the inconvenient 745.6494 relation which obtains when the standard 45 degrees of latitude is used.

Boiler Plate

The Massachusetts Board of Boiler Rules will hold its regular semiannual meeting on Thursday, Nov. 7, as provided by law, to consider changes in the boiler rules for that state. It would be a good time to propose a few changes in these rules regarding the requirements for boiler plate.

The rules specify that shells, drums and butt straps, shall be made of open-hearth "firebox" steel, and certain physical and chemical requirements are specified for this class of steel. There are two other grades of steel specified that may be used for other purposes, viz., "flange" and "extra soft" and the physical and chemical characteristics specified for the three overlap. Thus, if a steel of 55,000 to 62,000 pounds tensile strength has an elongation of 26 per cent. or more and a maximum of 0.03 per cent. phosphorus and 0.04 per cent. sulphur, and contains from 0.30 to 0.50 per cent. of manganese, it is either "flange" or "firebox," whichever the steel manufacturer chooses to stamp it. Similarly, a steel of 52,000 to 55,000 pounds tensile strength and 28 per cent. elongation and with the same maxima of phosphorus, sulphur and manganese is "firebox" or "extra soft," whichever the manufacturer prefers.

Since the chemical and physical qualities and not any process of manufacture determine the brand, why do not these qualities also determine the use that may be made of the material? A plate of either quality mentioned above, should be equally suitable under the law for use in a shell, regardless of the brand stamped upon it.

The law requires that the material that is to enter the shell of a boiler that is built for use in the state, be tested and the test certified to. Why should not these tests determine the fitness for use of the material tested? Considerable money has been spent by the steel and boiler manufacturers in restamping plates so that they might conform to the wording of the law after a boiler had been built and reached its destination. Surely such stamping added nothing to the safety of these boilers, and safety is the real and only object in the enforcement of the law.

Another point about boiler plate that might well be considered, is the present relation of its tensile strength to the pressure allowed on a boiler, which places a premium on the boiler manufacturer using material of the maximum allowed tensile strength. The rules allow a range of tensile strength for shell material from 52,000 to 62,000 pounds per square inch; consequently the boiler maker endeavors to obtain from the plate manufacturer the highest tensile strength material that he is willing to guarantee will not exceed the maximum allowed, because this will produce a boiler for a given pressure with a minimum of material and cost. As a consequence, practically all the plate that enters boilers built for the state of Massachusetts, has between 57,000 and 62,000 pounds per square inch tensile strength, this being as small a range as the steel manufacturer cares to keep within.

Almost all authorities agree that a lower range, say, 50,000 to 55,000 pounds would produce a material better suited to boiler purposes. If the method of fixing pressures on new boilers was arranged so that the actual tensile strength of the material for the shell did not enter the calculation, but that the lowest tensile strength allowed for this class of material would be used for this purpose, there would be no incentive to the boiler manufacturer to secure material of high tensile strength, and consequently boiler plate better suited for the purpose would be obtained.

One good reason for trying to keep near the lower part of the range allowed by the law is that the larger part of a plate is usually of considerably higher tensile strength than that of the coupon used for testing its strength. The test coupons are necessarily cut from the edges of the plate, where the impurities and the carbon content are less, due to segregation in the ingot, and consequently the tensile strength lower than for points further in from the edge. Where a coupon cut from the edge of a plate tests 55,000 pounds per square inch, the bulk of the plate is likely to range anywhere from 60,000 to 65,000 pounds per square inch in tensile strength.

Forehandedness

A man seriously ill calls a doctor or one in legal difficulties a lawyer, but very few are wise enough to seek doctor or lawyer for advice before they are actually in trouble. They forget that "an ounce of prevention is worth a pound of cure."

So it is with engineering. Many owners attempt to make changes in their plants or even erect new ones, without first obtaining expert advice. Then when they begin to have trouble, they call in the consulting engineer, and often much of the work, done wrong in the first place, has to be done over.

An example in mind is that of a small manufacturer who wanted to enlarge his roofing plant. Wishing to save money he decided not to call in an engineer to design it for him, but secured a millwright to do the work. As soon as the plant was started, trouble began from the tearing of the roofing. After wasting about a month trying to fix it, the owner sent for a consulting engineer. On checking up the speeds of the several drives, he found they were not running in unison, and it was necessary to redesign the plant, and order new sprockets, etc. So besides having to pay for the services of the expert, it cost the owner much lost time and added material.

Countless similar illustrations could be given of that foolish economy, which in the end costs many times what it saves. The man who will effect an ultimate saving and obtain the best economy is the one who does not begrudge a little additional first cost. Let the owner engage the engineer in the beginning; for he is the one best fitted by experience and education to do the work right, and will save his client many times his fee in future worry and expense. Happy is he who has this expert in the man in charge of his power plant; if he is a real engineer it is not necessary to go outside.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Indicator Diagrams from Engine and Steam Pipe

The two indicator diagrams, Fig. 1, were taken simultaneously, one with an indicator attached to the cylinder and the other with an indicator connected to the steam pipe supplying the engine. Both indicators were connected to the same reducing motion. The engine was a 20x42-in. Corliss, running at 74 r.p.m. connected by a 6-in. steam pipe 50 ft. long with two 72-in. by 18-ft. horizontal return-tubular boilers. A 60-lb. indicator spring was used. When these diagrams were taken the steam gage read 103 lb., but the steam line drawn by the indicator shows 101 lb. The two diagrams have been traced below each other so that the abscissæ of both correspond. The line diagram shows a sharp drop to 94 lb. as soon as the steam valve opens, but starts to rise just before the point of cutoff is reached.

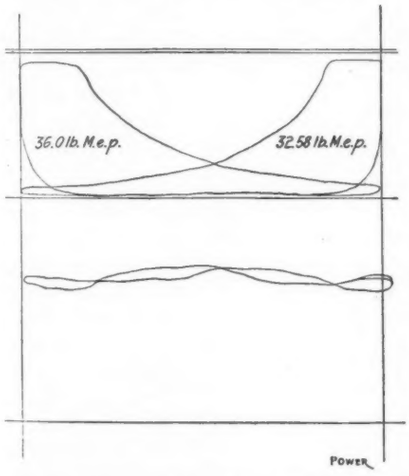


FIG. 1. INDICATOR DIAGRAMS FROM ENGINE AND ITS STEAM PIPE

The flow of steam from the boiler then raises the pressure line up to 110 lb., a return wave toward the boilers then causes the line pressure to fall to 96 lb. and the flow from the boilers then commences to raise the line pressure which reaches 102 lb. just as the steam valve at the opposite end opens. The line pressure now drops to 94.5 lb. but commences to rise near the point of cutoff until it reaches 108 lb. dropping, as a return wave starts to the boiler, to 97.5 lb. The two diagrams show a complete cycle of the pipe-line waves and their relation to the stroke of the engine. A number of other diagrams taken showed similar cyclic changes in the pipe-line pressures.

This succession of waves in the pipe line is a good illustration of why some pipe lines vibrate and likewise why a receiver close to the engine is advantageous. In this case the engine received 148 steam impulses per minute (two per revolution) and the waves in the pipe line have an approximate frequency of 296 per min., so that the live steam is "packed" into the cylinder at the beginning of the stroke by the wave impulse as the line steam wave is in phase with the speed of the engine. On some of the diagrams the pipe-line wave was slightly out of phase with the engine though in none of the diagrams taken did the steam valve open with the pipe-line pressure at a minimum.

The diagram showing the wave in the pipe line is distorted as the indicator was actuated by the crosshead of the engine which does not travel at a uniform speed. The average piston speed in this case is 8.63 ft. per second, the maximum being twice this, or 17.26 ft. per second, while the steam wave has a practically uniform velocity. To obtain the true shape of the wave it is necessary to re-plot it upon a base having a uniform time increment. The angular velocity of the engine crank is practically uniform; hence the distance the crosshead moves when the crank moves through any angle A or A_1 is equal to the algebraic sums of the versed sines of the angles through which the connecting-rod and the crank move. See Fig. 2, in which A, B and C represent the distance through which the crosshead moves while the crank moves through equal angles at different portions of its travel. Similar letters at the crank end show why this occurs.

As the crosshead starts toward the shaft its motion is the sum of the versed sines of the crank and connecting-rod angles ($a_1 + a_2 = a$). After the crank has passed over an angle of 90 deg., the motion of the crosshead equals $c_1 - c_2 = c$. In the diagram $a_1 = c_1$ and $a_2 = c_2$ and the crank positions are taken 60 deg. apart. By plotting the position of the crosshead at equal time intervals on the indicator-wave diagram, the abscissæ of the wave are obtained. The wave can then be replotted upon a base line having time abscissæ of equal values. This has been done in Fig. 3 in which the amplitude of the wave has been multiplied by five. Where the length of this wave is a harmonic of the distance between the boiler and the engine in the course in which the steam travels, or where it is a harmonic of the distance between sharp changes in the direction of flow, there will be periodic vibration set up in the pipe line.

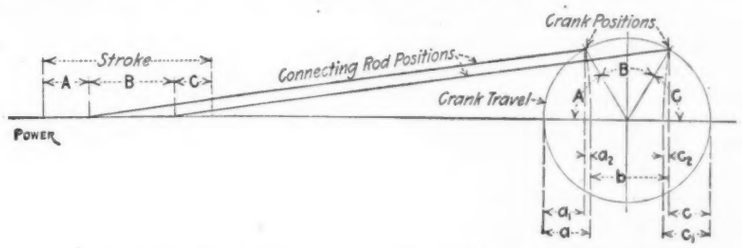


FIG. 2. SHOWING EFFECT OF ANGULARITY OF CONNECTING ROD

This wave action is entirely independent of the flow of the steam in the pipe line, that is, the wave travels through the steam at a velocity which is sufficient to cause an increase in pressure of 16 lb. The velocity of the wave can be computed by the following formula:

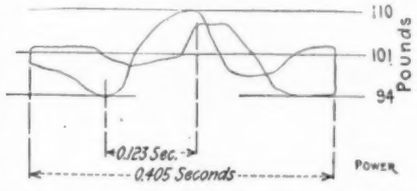


FIG. 3. AMPLITUDE OF WAVES IN STEAM PIPE DIAGRAM EXAGGERATED

$$v = \sqrt{182 \frac{Pg}{w}}$$

in which,

v = Velocity with which the wave travels in feet per second;

P = Pressure rise in pounds per square inch caused by the wave;

g = Gravity = 32.2;

w = Weight of the flowing substance in pounds per cubic foot;

182 = A constant.

For the case in hand the following numerical values can be assigned. P = 16 and w = 0.2681, the weight of a cubic

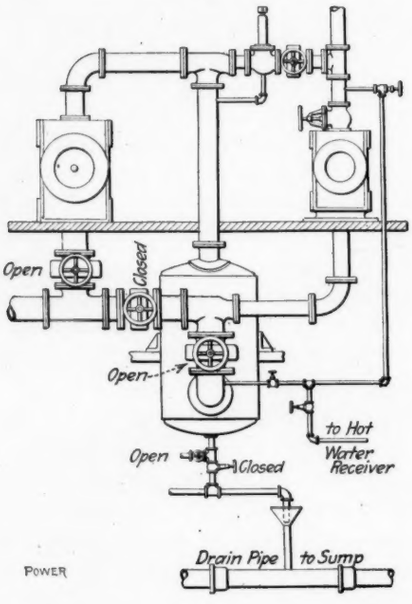
foot of steam at 103 lb. per sq.in. Substituting these values in the equation gives the value of v as 591.4 ft. per sec., which agrees quite closely with the periodic flow shown on the diagrams. The steam pipe is about 50 ft. long and the wave crests are less than 0.2 sec. apart, showing that the wave velocity exceeds 500 ft. per second. In some of the diagrams taken, the periodic rise in pressure was slightly less than that shown and in others slightly more. The diagram illustrated was selected because the wave was more clearly shown than in the others.

A. E. DIXON.

Cleveland, Ohio.

Rising Tide Fills Receiver

When the warming-up valve to the receiver of a cross-compound condensing engine was opened preparatory to starting the engine, a series of water-hammer blows gave warning that the receiver contained water, presumably above the height the warming-up pipe entered the receiver. The discovery led to considerable discussion as to how the water could have entered, and the possible damage which might have resulted if the engine had been started without preliminary warming, as sometimes happens when an unexpected call is made to meet a sudden increase of the load, or other emergency.



CONNECTION TO RECEIVER FILLED WITH TIDE WATER

All sources from which water might have entered the receiver were considered to determine the most likely one by a process of elimination. It was thought at first that the throttle might have leaked and the steam condensed in the receiver, or it might be the warming-up valve that was leaking. These possibilities were rejected as the receiver and the cylinders were found to be practically cold, and the receiver gage registered a partial vacuum of about 10 in., which was accounted for by the air and circulating pumps having been running for several hours.

It was then decided that this partial vacuum must have been responsible for the water being drawn into the receiver, though from what source was a problem yet to be solved, as the receiver drip emptied to the drain from an open-end pipe, the outlet being nearly 4 ft. above the water in the sump-well.

A few hours later the incoming tide solved the problem, it being unusually high and flooding the sump, it backed up in the drain until the end of the drip was submerged and the vacuum in the receiver caused it to be partially filled. Removing the nipple leading into the funnel made a recurrence of the flooding from this source impossible as there was no water seal as before.

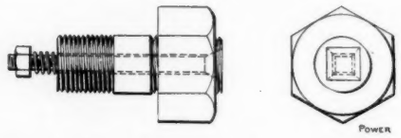
What damage might have resulted if the engine had been started with the receiver partially filled with water can only be conjectured, and I would appreciate learning the opinions of others as to what would have happened.

C. T. ADAMS.

Lynn, Mass.

Safely Follower Stud

The illustration shows an arrangement that has proved effective in preventing a broken stud and nut from dropping between the cylinder head and the moving piston.



CONSTRUCTION OF SAFETY FOLLOWER STUD

The center pin is a loose fit in the stud; and is made square, to prevent its turning around if the stud should break. The stub end should be drilled out as shown. The spring is to take the shock of a break, and also to hold the broken ends from vibrating with the motion of the piston.

R. DALE.

Joliet, Ill.

Insufficiency of Hydrostatic Boiler Tests

From time to time I, and probably many of the readers of POWER, have heard remarks which were not flattering to hydrostatic boiler testing. I have never had much faith in this means of determining the strength or condition of a boiler and some experiences which I have had recently have completely eliminated what little faith I may have possessed.

I am now convinced that cold-water pressure is satisfactory for determining the tightness of new work or for finding leaks which are known to exist, but as a means of determining the strength of an old boiler or what steam pressure a new boiler will stand, or whether or not an old boiler, or a new one for that matter, will be tight or stay tight under steam, I believe this so-called testing is about as useful as a fifth wheel on a wagon.

A short time ago I had occasion to put into commission a locomotive which had not been used for about three years. Shortly before it was laid up it had been retubed. Every "get-at-able" place was examined and sounded with a light hammer. Some corrosion was found, especially around the water leg; the tubes looked fairly good. I then connected up a hand pump and pumped up 200 lb. pressure, which was held while another inspection was made. Everything appeared to be satisfactory and the boiler was fired up, and, with the safety valve set at 125 lb., started to work on comparatively light hauling. The second day a tube developed a hole somewhere between the heads and leaked so badly that it was necessary to quit work and plug it. After this other tubes began to leak and it was ascertained that the tubes were in an unsafe condition.

On another occasion I did some experimenting on a 16-in. tubular steel automobile boiler having ½-in. tubes. This boiler appeared to be in good condition. Cold-water pressure was pumped up to 500 lb. on two occasions several days apart and the boiler was as tight as a bottle, but when 200 lb. steam pressure had been raised a superheater tube let go, and as the rear wheels were jacked up, the engine cut loose at a pace that drove all the "kids" and chickens to cover that were in the vicinity. With this boiler the steam after passing the throttle is carried down through a boiler tube to the firebox, then passes through a section of the superheater and up through another boiler tube, or so-called superheater tube, and then repeats this operation through another set of tubes and section of superheater. That makes it possible for any one of four tubes to fail and discharge the contents of the boiler into the engine with no means of shutting it off except possibly by putting the link motion in the central position. I cut the damaged tube out and found it corroded nearly through on top and also a small pit hole through it which had started the trouble.

A new tube was put in and a few trips made with the car, always expecting a little extra fun at any minute, of course, but nothing happened until one day the old "kettle" was fired up and when the gage showed about 50 lb. pressure the

bottom apparently dropped out of it. An examination showed water spurting from holes in four tubes.

These were plugged in approved automobile style and the experiments continued by pumping the pressure up to 500 lb. and holding it a short while for two or three times a day for several days, the object being to give the cold-water test all the chance there was to find any weak places which might exist. However, there were none for it remained as tight as a new boiler could possibly be. A slow fire was then started and the pressure brought up very gradually and when the gage showed about 225 lb., whish! went another tube. I am not exactly a novice in matters pertaining to steam and boiler inspection, and I should not recommend hydrostatic tests to reveal what an engineer wants to know of the condition of a boiler.

H. L. STRONG.

Portland, Maine.

Another Experience with the Central Station

Recently I visited an engineer and he confessed his fear of the central station, but knew he could compete with it if the owner would coöperate with him.

On examination I became convinced that the solicitor was figuring him out of a job. The two boilers were fed by injectors. The engine which developed about 80 hp. exhausted direct to the atmosphere. Live steam was used for drying purposes and also for heating when it was required. The engineer said he had been "turned down" every time he mentioned using the exhaust steam. The owner came into the engine room and I was introduced to him and soon led the conversation to central-station service, which he said he contemplated using. I inquired as to the gain he expected and he replied that it would do away with one boiler and the other could be fired with sawdust and shavings (a byproduct) and he could sell the butts and slabs for wood; that he would not have to pay an engineer, and would save the price of about 125 tons of coal a year which was then required.

I pointed out to him that by using valves and piping he could heat the kiln and factory with the exhaust steam, and with a pump and heater to use the returns for boiler feed, could save the coal bill just the same; that he would have to pay an oiler to look after shafting, and that I thought he could do without one boiler.

"Why!" said he, "the engineer has been telling me that, too!"

I referred him to an engineer whose specialty is steam heating and got him to promise he would consult him. I thought little more about it, until this owner wrote me saying the new system was installed, that they were using but one boiler through the summer and might be able to run with one all the time.

V. C. WOOD.

Copenhagen, N. Y.

Common Trouble of Siphon Condensers

Whenever a valve breaks in a water pump supplying a siphon condenser, it usually follows the water up and lodges between the cone and the casting, as shown at *A*, Fig. 1. Impurities, small eels, clam shells, etc., have also been found blocking the space as to seriously impair the vacuum.

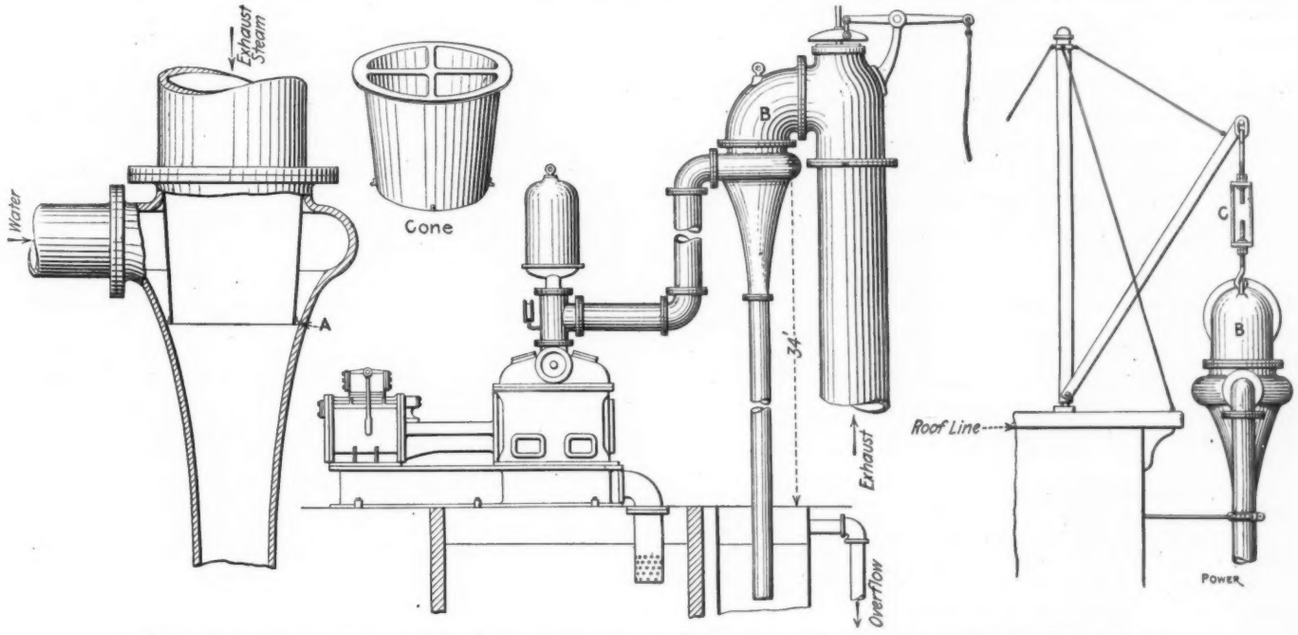
Under this condition of stoppage, a pump becomes overworked to a point above safety. Generally 3 or 4 lb. is registered on the pump gage, under normal conditions, but when the opening is badly plugged, 16 to 18 and sometimes 20 lb. is required to force water through the condenser, and consequently the strain is plainly visible on the pump.

In some places running but 12 hours, the day can pass without interruption, and at 6 p.m. the condenser may be easily cleaned. This is done by removing the bolts and raising the elbow *B*, Fig. 2, then pulling up the cone shown in Fig. 1. If the roof of the mill or shop is higher than the condenser, the elbow can be raised with block and tackle. If lower, the arrangement shown in Fig. 3 will do very well. The elbow is lifted by turning the buckle *C* with a short bar; it is then swung aside. The cleaning consists of hauling the cone out with both hands. At the instant it is raised, the obstructions will be seen chasing down the water-leg and into the well. The cone may be set aside and, if any stuff remains it can be brushed away with the hand. Extreme care, however, should be exercised in replacing gaskets and making joints. Gaskets should be painted with graphited oil and then the bolts tightened cautiously.

An improvement has been made of late years which facilitates this whole operation. It consists of a geared mechanism on the outside of the elbow. And by turning a wheel the cone is raised. The cleaning can thus be performed with greater ease and at any time when running.

LUKE MARIER.

Fall River, Mass.



PARTS AND ARRANGEMENT OF SYPHON CONDENSER AND MEANS FOR REMOVING ELBOW TO CLEAN CONE

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

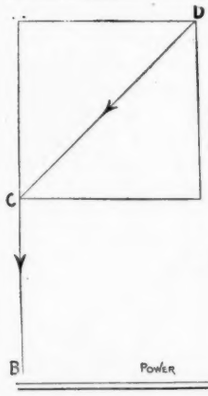
Test of Axial Flow Centrifugal Fan

In the June 25 issue a letter appeared by me relative to an axial flow turbine fan. Those interested in this fan will probably find the following test results of value:

TEST OF AXIAL FLOW CENTRIFUGAL FAN

Intake Side						Outlet Side							Input					
Stat. suction at center	Velocity pressure at center	Air velocity from reading	Average velocity over pipe	Capacity cu.ft. min.	Hp. air	Velocity pressure	Air velocity from reading from reading	Hp. air	R.p.m.	Rim speed	Ratio $\frac{\text{vel. rim}}{\text{vel. entry}}$	Ratio $\frac{\text{vel. rim}}{\text{vel. delivery}}$	Volts	Amps.	E.Hp.	B. Hp.	Mech. Efficiency as an Exhauster, Per cent.	Mech. Efficiency as a Blower, Per cent.
$\frac{1}{2}$	$\frac{1}{16}$	1394	1270	725	0.0142	$\frac{1}{2}$	1971	0.0284	1100	2735	1.96	1.39	46	1.4	0.087	0.057	25	50
$\frac{9}{16}$	$\frac{1}{16}$	1707	1560	890	0.0262		2415	0.0525	1320	3283	1.92	1.36	58	1.65	0.128	0.084	$31\frac{1}{2}$	$62\frac{1}{2}$
$\frac{5}{8}$	$\frac{1}{16}$	1971	1790	1020	0.040		2789	0.08	1540	3830	1.95	1.38	75	2.05	0.206	0.136	$29\frac{1}{2}$	59
$\frac{11}{16}$	$\frac{1}{16}$	2204	2000	1140	0.056		3118	0.112	1760	4377	1.99	1.41	90	2.35	0.284	0.188	$29\frac{1}{2}$	$59\frac{1}{2}$
$\frac{3}{4}$	$\frac{1}{16}$	2415	2200	1250	0.0745		3417	0.148	*1870	....	....	....	102	2.7	0.369	0.243	$30\frac{1}{4}$	$61\frac{1}{2}$
$\frac{7}{8}$	$\frac{1}{16}$	2590	2350	1340	0.0925		3690	0.184	*1980	....	....	....	111	2.8	0.417	0.275	$33\frac{1}{2}$	67
																	Average 30%	60%

*These r.p.m. are evidently too low.



PARALELLOGRAM OF FORCES

AB = Rim velocity of wheel

AC = ½AB = speed of vel. center

AD = AC = ½AB = vel. of entry of axial flow.

CD = resultant or velocity of delivery = $\frac{AB}{1.414}$ since paralellogram is a square.

Description

Wheel 9½ in. diameter. Inlet pipe 10¼ in. diameter, six ft. long with flared mouth giving effective inlet equal to that of wheel (62 sq.in.) Area pipe 82 sq.in. or 0.57 sq.ft. Average velocities in pipe taken as 0.91 of that of center. Readings taken with two Pitot tubes at center of pipe, the static and dynamic tubes being connected to each leg of a *u*-tube, thus reading velocity pressure. Readings are in inches per sq.in. of water.

Motor ¼ hp. 110 volts series wound, direct connected with flexible coupling. Prony brake test showed motor efficiency to be 66 per cent.

Mech. efficiency of wheel given above includes two babbitt bearings. At 5500 ft. rim speed it is silent running.

About 5200 ft. was the highest peripheral speed obtainable with the motor used; at 8000 ft., which is not excessive with this type of wheel, the air velocities on each side of the wheel would correspond to 1 in. and 2 in. of water, respectively, while handling 2000 cu.ft. of air per min.

C. O. BRETHERICK.

Seattle, Wash.

Priming Pipes

R. T. Rezniem's letter in the Sept. 10 issue describes a practical arrangement for priming a pump. I would have appreciated a longer treatment of the subject than given by Mr. Rezniem.

When pumping liquids from a well or tank below the pump, the liquid will run back into the well when the pump is stopped if air leaks into the suction or the discharge chambers. If the discharge valves seat improperly the liquid will leak through them and also through the suction valves if they are badly worn or cracked. Air then fills the space formerly occupied by the water.

If there is a column of water in the discharge pipe and the discharge valve is opened the discharge chamber is immediately filled with water and the pressure due to the head holds the discharge valves so tight against their seats that no water gets to the pistons and suction valves to keep them water-sealed and air-tight. The piston creates a very low vacuum by its displacement in moving away from the end of the cylinder; it will also compress the air in front of it against the discharge valves. The air must be compressed to a pressure equal to and above that in the discharge pipe to open the discharge valves, but it will follow the piston and expand and again be compressed as the piston moves back and forth. This low air pressure in the cylinder only aids to keep the suction valves closed and the pump becomes air-bound.

The water is surging back and forth through the priming pipe as it is discharged from the cylinder, but with the air cock open, the air is liberated. Water passing from the discharge pipe to the suction-valve chamber would keep the valves sealed air-tight and thus exclude the air until a sufficient vacuum had been created in the pump for the atmospheric pressure on the water to force the latei up into the pump. Then the liquid, being a solid body would, when pushed by the piston, force open the discharge valves and the pump work properly.

R. A. CULTRA.

Cambridge, Mass.

Power for Salt Works

The article on this subject by Mr. Saxe, page 219, Aug. 13 issue, is very interesting and recalls my experiences with such plants.

From his descriptions of the grainers, I believe he could have returned the condensation to the boilers without such a great temperature loss. Three or four per cent. CO_2 is not unusual in such plants, in fact, it is quite common in plants where the firemen are greatly underpaid. If Mr. Saxe could have instituted a bonus system for the firemen it might have brought excellent results.

Mr. Saxe said induced draft would have been more economical and produce less smoke than the forced draft used. In my opinion forced draft would prove better than induced draft. With forced draft there should be a higher percentage of CO_2 than with induced, and a lower stack temperature, if the inside of the boiler is free from scale and the outside free from soot.

Plant designers seem to have a ten-

dency to provide too large a grate area. Often I have found the grates so large that it was only possible to burn about 10 lb. of coal per hour per square foot of grate. Mr. Saxe could have dispensed with a number of boilers, burned more coal per hour under those in use and obtained a high percentage of CO_2, and undoubtedly a higher efficiency, if the settings were good, and the boilers clean inside and out.

Another factor in boiler efficiency often overlooked is the relation between the chief engineer and the fire-room force. The fireman should be taken into the confidence of the chief. Let the chief teach the fireman how to fire, then pay him what he is worth.

G. F. DUEMLER.

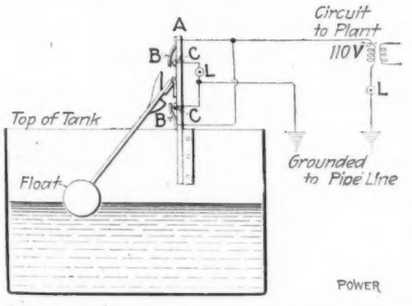
Philadelphia, Penn.

Water Tank Signal System

The recent letters and illustrations on this subject have interested me very much. We have had a signal system in operation for several years that has given good service.

The present water-system reservoir is not high enough to supply a section of the city, so a pump was placed in the power station and takes water from the mains and forces it to a tank about 1½ miles distant and high enough to supply the high section. As the pump was only required to run a part of the time a signal system was required to tell when the tank was empty or full. The system shown was worked out and installed.

A is a piece of timber placed in the tank in a vertical position. *BB* are strips of spring brass, and are forced in contact with plates *CC* by the float lever. A single wire is strung from one side of a 110-volt circuit at the station and connected as shown. The other side is connected near the pump to the discharge line with a 110-volt 16-cp. lamp in circuit, and contacts *CC* are connected to the same pipe line at the tank with a lamp near the upper contact.



MECHANISM OF HIGH- AND LOW-WATER ALARM FOR TANK

When the tank is almost empty the float drops and the lever causes the lower spring *B* to make the contact with *C*, completing the circuit, and the lamp *L* will burn at full brilliancy. When the tank is full the float lever causes the upper spring *B* to make contact and complete the circuit through the lamps *L* and *L* and the lamps burn at half brilliancy.

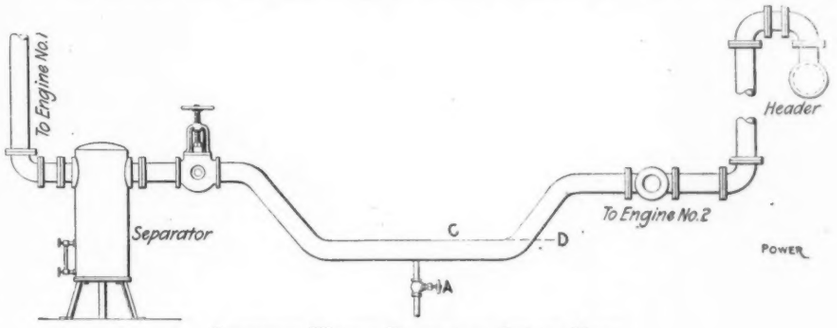
The float lever is made fast to piece *A* by a hinge. The system has given very little trouble and has the advantage of giving two different signals without switches.

C. E. NIGH.

Morgantown, W. Va.

Faulty Pipe Design

J. W. Parker's letter on this subject in the July 30 issue reminds me of a piping layout arranged similar to that shown below. The drop in the pipe was put in to avoid a large exhaust main. The drain *A* had not been connected to a trap. The installation was a new one and engine No. 1 had not yet been started, and the stop valve at the separator had not yet been opened. Engine No. 2 had been running for a few days, and the pipe was therefore full of condensation up to the stop valve. Apparently no water-hammer had occurred to warn anybody, although the straight run at *C* must have measured at least 10 ft.



SHOWING WATER POCKET IN STEAM MAIN

One night, steamfitters were working near the drain and for some reason opened the valve. Violent water-hammer occurred soon after the valve had been opened. I should say it occurred when the water reached the level *D*, and began to run back from the stop valve shown. The pipe was broken and twisted, and the separator upset. One steamfitter was badly burned; the others escaped through the nearest exit.

R. MCLAREN.

Govan, Sask., Canada.

Water-Sealed Flue-Gas Collector

In the water-sealed flue-gas collector shown on page 269 of the Aug. 20 issue, it would appear that the collecting cylinder is a little too large for the water holder. With the various dimensions given the water in the apparatus now stands almost in equilibrium, which means that all the lower cone is of no value as a part of the collecting apparatus. An overflow vent might be placed about 5 in. from the bottom of the outer vessel, which would allow the water to come finally to rest at that level, making all the inside of the collector available.

Mr. Barnum does not speak of one point I have found important. The water used in a collecting system of this sort should not be fresh. Water has the power of absorbing nearly its own volume of CO_2, and if fresh water is used in an apparatus of this sort, when the sample is to be tested later for CO_2, the results are invariably low. The reverse of this also seems unfortunately true, i.e., if water saturated with CO_2 is used to collect gas, which is low in CO_2, the water will give up a little of the carbon dioxide to the collected gas.

However, most flue-gas work is done on gases which do not vary greatly, and I have found it gave the best results to use the same water for collecting the gases over and over, until it became too physically dirty to use any longer. This applies chiefly when analyzing for CO_2, but slightly to CO, the latter being comparatively insoluble. The power engineer rarely is concerned with the oxides of sulphur. If it becomes necessary to investigate their proportions, he may as well give up any idea of attempting to collect the sample over water; some absorption process must be used.

DONALD M. LIDDELL.

Elizabeth, N. J.

Confessions of an Engineer

Mr. Rogers' article, "Confessions of an Engineer" in the Aug. 20 issue reminds me of an experience I had with a superintendent.

The building was an old one and badly in need of repairs and the shafting was not unlike that mentioned in the article. It had not been lined up since it was installed. One could stand anywhere in the room and see that it was way out of line. We used automatic grease stick lubricators and some of the bearings ran so warm that a stick would not last half a day.

The superintendent believed in letting

things run as long as they would, regardless of their conditions, and when I asked for a helper to align the shafting he replied that I could not do anything with it, that the building had settled and as long as it ran, not to bother with it.

I was not satisfied to think that way, so tackled the job alone that afternoon, using only a spirit level and line. In some places I had to raise the shaft about ⅜ in. and nearly as much sidewise in other places. The work was not perfect, but it helped so much that I was not bothered with hot bearings during the remaining six months I stayed there.

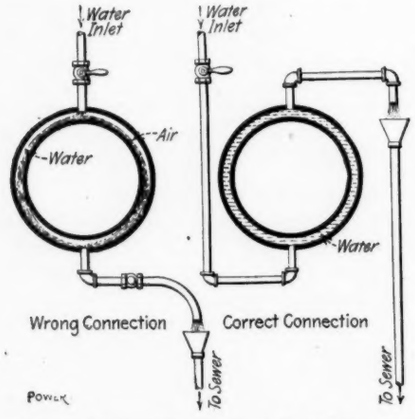
Another experience was with a millwright when I wanted to change the belt on a small dynamo. The one in use was about worn out and I had selected as its successor a belt that had been run in a dry place with no attention and had cracked considerably. He claimed it would never do the work and I might as well get a new belt first as last. Nevertheless, I had him make the joint on Saturday and I left the clamps on until Sunday and gave it a coat of good dressing. I followed this up with a lighter coat when the belt had absorbed the previous ones, and was never bothered with the belt not doing its work properly. He was quite surprised at the condition of the belt when we removed it a few months later, having no more use for that dynamo.

G. B. LONGSTREET.

Somerville, Mass.

Value of a Sketch

Soon after arriving in this country, I secured a position as general mechanic with a Canadian railroad. I could not speak English very well so experienced trouble in connection with my work.



JACKET CONNECTIONS TO AIR COMPRESSOR

As mentioned in the editorial and also Mr. Dixon's letter under the above headings in the Aug. 13 issue, I found my ability to sketch very valuable in conveying ideas.

On one occasion I was sent to correct the trouble in an air-compressor cylinder which was always running hot. The foreman was very upset over the case as a man had been working on it for three days and left the job with the cylinder in the condition he found it. The water connections were wrong and after a half hour's work I changed them and the trouble ceased. I had considerable difficulty in explaining how the job was done, so I drew the sketch shown so he could understand what I tried to explain. He was so pleased he told me to take the rest of the week off and go fishing.

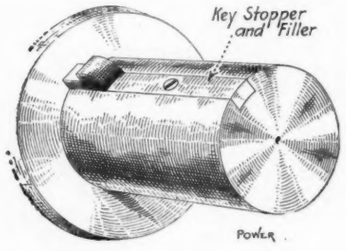
J. G. KOPPEL.

Montreal, Que., Canada.

Safety Pin for Flywheel Keys

A. L. Haas' letter under this heading in the Sept. 3 issue interests me. While the job illustrated and described will undoubtedly prevent the key from coming out, it does not appeal to me as a good job. It is, as many old mechanics express it, "painful to the eye." It looks clumsy and awkward, but appearance is not always considered so long as solidity is obtained.

A solid job can often be had and at the same time the appearance made desirable. It does not seem reasonable to drill a comparatively large hole clear through the end of a shaft, and fit a correspondingly large taper pin in it to prevent the key from backing out. A better plan would be to fill the unused part of the keyseat with a neatly fitted false key, as shown. This will at once act as a stopper and filler, and preserve the symmetry of the shaft end. I have seen this done on large engines, and no trouble was ever experienced from the key coming out.



A "MECHANICAL" WAY OF FITTING A FALSE KEY

CHARLES J. MASON.

Scranton, Penn.

Paint for Coils

In answer to B. A. Warren, who asks for a paint for coils in the Aug. 13 issue, I would advise that a good, serviceable paint for coils used in refrigerating work, smokestacks or any iron work can be made from 5 gal. coal tar, 1½ gal. paraffin oil and 3 pints of Japan dryer. Mix and stir until the coal tar has cut the paraffin. This will take from 5 to 10 min. This paint will outlast asphaltum when it comes in contact with water or brine.

E. F. TRACY.

Mechanicsville, N. Y.

Shorter Cutoff Greater Load

Mr. Stewart, writing on this subject in the Sept. 10 issue, has not made his argument as clear as he might, although he is correct in most of his statements.

Relative to a shorter cutoff in the low-pressure cylinder causing a loop in the high-pressure diagram, he might get a negative loop in this diagram on no load and still have the valves set so as to cut out the loop at half load. All compound engines are not governed alike, but as Mr. Stewart says, the load can be shifted from one cylinder to the other by changing the low-pressure cutoff, but he neglected to state that the power of the engine remained the same. The indicator will show that a change in the receiver pressure, due to manipulation of the low-pressure cutoff will change materially the amount of work done in each cylinder, although it will leave the total work done by the two cylinders the same as before.

So if the receiver pressure is lowered by making the low-pressure cutoff later the back pressure obviously will be less in the high-pressure cylinder and the mean effective pressure will be more, so that the high-pressure cylinder will now be doing more work. Since the total work done remains the same, the low-pressure cylinder in spite of its later cutoff will now be doing less work.

Conversely if the receiver pressure is increased by making the low-pressure cutoff earlier, less work will be done in the high- and more in the low-pressure cylinder. To change the power of the engine the high-pressure cutoff must be changed, while to distribute the work in the two cylinders the low-pressure cutoff must be altered. The governing mechanism of many compound engines operates on the high-pressure cutoff, but this is open to the serious objection, that when the governor makes the high-pressure cutoff later in responding to an increase of load, the larger share of the work will be done in the low-pressure cylinder, and when the governor causes cutoff in the high-pressure cylinder earlier, the larger share of the work will be done in the high-pressure cylinder and at very early cutoffs the low-pressure side may then actually be a drag on the engine. To overcome this many compound engines have the governor operate the cutoff on both cylinders. Then the governor operates the high-pressure cutoff to change the power of the engine and the low-pressure cutoff to equalize the work done in the cylinders.

C. F. ROBINSON.

Iowa City, Iowa.

Making a Concrete Engine Foundation

By H. L. Strong

Directions for making the form and proper mixture for a concrete engine foundation. Engineers have to do this work sometimes and the article is worth filing for such an emergency.

Matters pertaining to concrete may not be considered as within the province of the steam engineer, but it is well to be able to do a little concrete work as occasion may require.

For example, the boss inquires as to whether the engineer can install a new direct-connected set, and also the material he considers best for the foundation. Concrete is recommended as it is generally used and costs about $5 per yard in small quantities, and makes a very firm foundation, whereas brickwork costs from $1 to $2 more and is no better, if it is as good. The employer is pleased to have a man who can care for the installation of an engine from the foundation up. He can give this man the foundation plans and it may be completed when the engine arrives from the makers.

As there is usually a heavy concrete floor at the place where the foundation is to be, a few steel bars like that shown in Fig. 1 should be made. A couple of 6- or 8-lb. striking hammers, and four men should now be put to work cutting out the concrete floor, one man holding a "point" and another striking it. After a hole is started the floor may be torn out quite rapidly. When the hole is cut to the required size it should be trimmed up to the lines, so as to make a smooth job of patching in after the foundation is completed. These tools are also used for cutting mass concrete, and should be kept sharp for easy and rapid work. Made round as shown, they are much superior to the square or chisel-shaped point com-

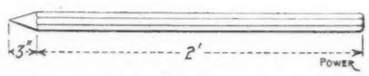


FIG. 1. POINT FOR CUTTING CONCRETE

monly used, as there is less breakage and sharpening needed and they penetrate much easier.

Having cut out the floor and excavated to rock, it is now in order to compute the cubical volume of the foundation, which it will be assumed is 300 cu.ft. This divided by 27 gives 11⅑ cu.yd. The mixture should consist of, in masonry parlance, 1, 2½ and 5, which means 1 part cement, 2½ parts sand and 5 parts crushed stone. As the cement and sand make about mortar enough to fill the voids in the stone, 12 yd. of stone will do (allow something for loss in handling, etc.) Half as much sand, or 6 yd., is needed and as this mixture takes about 1½ bbl. of cement per yard, 18 bbl. are required. If the cement comes in bags, four bags equal a barrel and 18 × 4 = 72 bags. A bag holds 1 cu.ft.

A mixture of 1, 3 and 6 also makes good concrete for mass work and does not take quite as much cement. For very particular work 1, 2½ and 4 or 1, 2 and 4 is frequently used. "Fine stuff" for finished surfaces is made of 1 part cement and 2 parts sand, the sand should not be too coarse. As the cement fills the voids in the sand, a yard of sand makes about a yard of mortar. The engineer should now estimate the material needed by weight, as that is the way it is usually sold. Sand weighs 100 lb. per cu.ft.; therefore

$$27 \times 100 \times 6 = 16{,}200 \text{ lb.}$$

or 8⅛ tons is required. Crushed stone weighs 90 lb. per cu.ft., and

$$27 \times 90 \times 12 = 29{,}160 \text{ lb.}$$

or 14.58 tons will be needed.

A finished surface is not necessary on the foundation as it will be out of sight; otherwise the form should be made of matched and planed spruce boards. Good square-edged boards will do and they are cheaper. The number of square feet of 1-in. boards and linear feet of 2x4-in. joist required to build the form should be determined. Add enough to this amount to construct a 10x10-ft. double-boarded platform for a mixing-board.

Old or stale cement should not be used. To test for its quality, mix up a sample and if it does not set firm enough in 24 hours to be handled without crumbling, and so a knife blade cannot be pushed into it, it is not fresh. The test is rather crude, but after a little experience one can judge correctly as to whether or not concrete is setting properly.

Cement sets best in a moderate temperature. Too much heat is not desirable and any cement sets slow in cold weather. Clear, "sharp" sand and clean, crushed stone which has been properly screened should be demanded, as good concrete cannot be made with poor cement or soft, dirty sand and "any old thing" for stone. Soft sand or even a small amount of earth in the material will prevent the cement from setting properly. The writer has seen concrete made from such material that could be kicked to pieces after it had set several days. Such concrete is worthless for any purpose and is the cause of much adverse criticism of concrete in general. If proper materials were always used in making concrete, there would be few cases of its failing to prove satisfactory, unless from improperly designed structures or removing the false work too soon.

Make up the broad sides of the form by cutting the 2x4-in. joists into lengths an inch or two shorter than the height of the foundation. There should be enough pieces to allow for spacing them 18 or 20 in. on centers. To do this, lay them down on edge, mark the spacing on a board and nail it to the 2x4 joists with 6-penny wire nails, and continue to nail

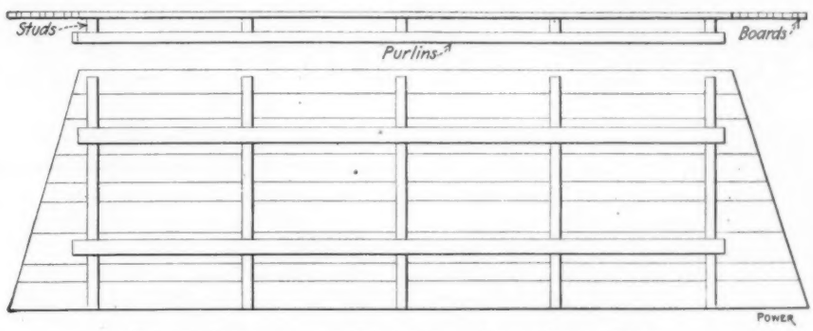


FIG. 2. DETAILS OF A SIDE FOR A FOUNDATION FORM

on boards until the required width is attained. The ends of the boards are then marked and sawed to the required length, and as the bottom of the foundation is wider than the top, the cut has to be made at an angle. Make up these pieces until there are enough to form a box just the size and shape of the foundation, the boards on the top extending an inch or two above the joists. Now cut two pieces of 2x4 for each side of the box to be used for "purlins" and nail them across the studding. One of these sides is shown in Fig. 2. The pieces are then assembled in their proper places and the form is squared. The next step is to solidly brace the form from the outside by cutting 2x4 shores which fit tightly between the purlins and boards laid against the sides of the excavation.

The form can now be lined up by driving wooden wedges around the shores. It should next be accurately leveled, as it is both troublesome and expensive to

patch up defects in the concrete due to inaccuracy of the form.

Fasten pieces of board across the top of the form, on the lines of the foundation or holding down bolts. Carefully locate the positions of the bolts on the boards and bore holes in the boards just large enough to take the bolts. Bolts of the required diameter and 2 or 3 ft. long, having square heads and heavy washers or a few inches of the end turned at right angles are then hung through the holes, the nuts on them being screwed down enough to keep the ends above the false work an amount equal to the thickness of the nuts, washers, bedplate and to allow for grouting under the bedplate. Pieces of pipe, larger than the bolts and a foot or more long, should be slipped over the bolts and fastened up under the boards or template instead of cementing the whole length of the bolts in solid, as this will prevent trouble if the bolts do not enter fairly when the machine is set in place.

Next make a bottomless box 3 ft. by 3 ft. 4 in. by 1 ft. = 10 cu.ft. This box is placed on the mixing board and filled level full of sand and then lifted up, leaving the sand on the board. One barrel (4 bags) of cement is then dumped on the sand and the two thoroughly mixed together while dry by turning over two or three times with square-end shovels. The mixture is then roughly formed into a basin and three or four pails of water poured in and the dry material worked over into the water, and spread out. The bottomless box is then filled twice with crushed stone which is dumped on the mortar and the whole mixed thoroughly by shoveling it over three or four times and adding water enough to make it slightly sloppy or so that it will run off a shovel or out of a wheelbarrow without much scraping. The old method of barely wetting the mixture and then pounding it into the form is obsolete, as concrete mixed in that manner cannot be as dense or strong as by the new method of running it into the form. An excess of water is also detrimental as it washes the cement out of the mixture. Dump the mixture into the form, but remove the inside braces as the form fills. A flat spade is also worked up and down next to the form to push the stone back and let the mortar fill in smoothly along the face. This is especially necessary if a smooth surface is desired.

This operation is continued until the form is filled to within 2 or 3 in. of the top. The concrete is then allowed to set enough so the template can be removed without throwing the bolts out of line. Some 1-and-2 mortar is then prepared and the form filled full and "struck off" by working a straight-edge across the top, after which the surface should be smoothed with a wooden or steel "float" (large, flat trowel). In three or four days the false work can be removed. The space around the foundation is then filled in, the filling tamped down solid, and the floor patched in. It is well to allow the block to set for about a week before setting the engine on it. After the engine is on the block, it may be shimmed up with pieces of sheet iron or wedges, if this is necessary to level it.

Now mix some cement and make a dam 2 or 3 in. high all around the base, after which mix some "half and half" grout and pour it in around the base. Work the space full with pieces of thin, flat iron. Pouring is continued until the dam is also full, which gives head enough to overcome the shrinkage as the grout sets. Before the grout and dam have set hard the cement is neatly trimmed off with a trowel, and in three or four days the nuts on the holding-down bolts may be set up.

Design of Surface Condensers

By W. H. Booth

A suggestion that the tubes of a condenser be spaced wide in proportion to their nearness to the cylinder, the passageway for the steam diminishing as the steam diminishes in quantity. This would tend to produce a uniform vacuum throughout the condenser.

In designing surface condensers the main object generally seems to be to put the largest possible surface in the direct path of the steam irrespective of any consideration for the passage of steam beyond the first row of tubes. About the only concession in this direction is the omission of a group of tubes so as to produce a gap of V-form in the general mass of tubes. These gaps serve to present more tube surface on the one hand, while on the other they provide more passageways by which the steam can reach the interior of the nest of tubes.

Yet the modern steam turbine is found to be so arranged as to provide a pipeway to such a blocked condenser, the full diameter of the turbine rotor. If so great a diameter of this detail be desirable, it is not consistent that the passageway between the condenser tubes should be so much reduced. Experience also points to the error, for the vacuum differs by as much as an inch between the steam entry and the base of the condenser. Except for considerations of cost and space, there is no reason why a condenser should not be many times its ordinary cubical capacity with widely spaced tubes dispersed throughout the space. If steam be regarded in the light of the kinetic theory of gases, the molecules of which are darting about at a velocity of perhaps 1800 feet per second, is it not obvious that an extra few feet of length or breadth will have little effect upon the time required by each molecule to find its way to a cold tube surface? The ideal condition would be that each molecule touch a cold surface which would kill its self-translatory activity and drop it to the condenser base as so much dead water.

Every tube carrying a stream of water possesses a certain capacity for absorbing heat from the molecules which pass upon it. But if too many molecules strike the tube in a given time the tube surface will become hot for there is a limit to the rate of heat conduction through the walls of the tube. From the hot surface the steam molecules will rebound without having given up much of their heat energy and in the rebound they will strike against the oncoming stream of molecules from the cylinder, driving these back to some extent or at least hindering their flow. Undoubtedly the major portion of the work of a surface condenser is accomplished in the first row of tubes, and the result is this reflex action of many of the molecules which ought never to have hit any of the first row of tubes but should have passed forward to the second, third and subsequent rows. What seems to be needful is that only one tube in three of the first row should remain, and this tube should be fitted with a light casing of thin metal with lips placed vertically toward the flow of steam. In cross-section this casing would resemble the letter *A* with an open top. The tube would occupy the lower division of the letter which would rest upon the tube by means of an occasional cross bar. The top slot would have a breadth of, say, a fourth or a third of the tube diameter, thus reducing the number of molecules which reach the tube surface and therefore keeping down the temperature of that surface so that it ceases to be a surface from which molecules can rebound with any great velocity. In this way the amount of steam dealt with by the first row of tubes is reduced to about one-ninth or one-twelfth of the old arrangement. The next row of tubes would be similarly spaced and fitted and so also the third or even the fourth rows; beyond that the spacing would become closer and

the covers made with wider slots or even omitted, and so on to the base of the condenser, where the tubes would be closely spaced.

The provision for wider spacing is particularly desirable where the contraflow principle is employed, for if this is fulfilled the first tubes touched by the steam are the hottest and create the maximum of reaction. Since the temperature of saturated steam is fixed for any given pressure there can be no temperature difference without a pressure difference, and the contraflow principle presupposes such a pressure difference and indicates a loss of vacuum in the cylinder. It seems more rational that the first rows of tubes should be as cold as possible and that the last rows should be cold also. Any marked rise of temperature of the circulating water should be in the middle tubes where most of the steam has disappeared and the passageway is ample for what is left. There appears to be no difficulty in designing a condenser on these lines to provide for intercepting the water condensed in each division so that it shall not drip on the tubes below. Summarized briefly, the suggestion is that the spacing of the tubes should be wide in proportion to their nearness to the cylinder and that the passageway should diminish as the steam diminishes in quantity.

If a 2-inch vacuum is attainable under certain conditions and there is an absolute pressure of 3 inches at the top row of tubes, the steam volume per unit weight will be 50 per cent. greater at the lower tubes and should be provided for accordingly. But such a loss as 1 inch is what the suggested design aims to avoid by the provision of ample space for the passage of the steam. In practice it appears probable that such a design would culminate in condensers having a maximum of passage area transverse to the path of the steam, so that the condenser would be long and wide in order that the first rows of tubes should still possess considerable surface even though the tubes were not allowed to exceed a given fraction of their full duty, as provided for by the cover sheds.

Steam is perhaps best distributed over the upper row of tubes by providing for its entry above the tubes and parallel with them in a wedge-shaped chamber extending the length of the tubes. If admitted to the central part of the area of the upper row of tubes it should be by a tapering entrance or a reversed hopper springing from both ends of the tubes and fitted with diverging plates to spread the steam throughout the length of the upper row. Baffle plates across the path of the steam are bad practice and serve to cause obstruction. It is better to guide the steam than to compel it to spread by the interposition of a baffle.

The ideal condenser is, of course, a large open chamber with cold walls, into which steam is admitted and speedily finds one of the six cold walls. A large cubical room with about 45 feet square on each wall would give only the 12,000 feet of surface regarded as necessary for a tubular condenser occupying a mere fraction of this volume. Even cellular, flat internal walls would diminish the volume very little and it is not easy to escape from the tubular design. But this has been overdone to the extent that tubes have been spaced too closely to permit the vacuum formed at the base to act upon the cylinder.

The Sulphur Contents of Petroleum

The Bureau of Mines has just issued a bulletin describing different "Methods of Determining the Sulphur Contents of Fuels, Especially Petroleum Products," by Irving C. Allen and I. W. Robertson. The paper describes in detail eight methods of sulphur determination and the conclusion of the authors is that the best method is that of burning the fuel in a calorimetric bomb with pure oxygen under a pressure of 30 to 40 atmospheres. This method is accurate, practicable and rapid and is recommended in preference to all of the other methods described. The calorimetric determination if desired can be made at the same time. Copies of the paper can be obtained by addressing the Bureau of Mines at Washington, D. C.

Automatically Operated Valves

The Golden-Anderson Valve Specialty Co., Pittsburgh, Penn., manufactures a line of automatically operated valves. That shown in Fig. 1 is a double-cushioned, triple-acting, nonreturn valve, designed to automatically protect boilers in case of a bursting tube. It will also act as a safety stop and prevent steam entering a cold boiler.

The dashpot area of the upper part of the body insures a cushion in the opening and closing of the valves and the alignment with the seat regardless of position.

The dashpot *A* is attached to the valve spindle; the outside of the dashpot acts as a bronze cylinder and is held firmly in place by cap bolts.

A bypass, admitting full boiler pressure above and beneath the inside dashpot, shown at *C*, balances it. The steam connection from the main steam line to the automatic pilot valve holds it closed, because of the large area above it.

Between the two dashpots is a steam-

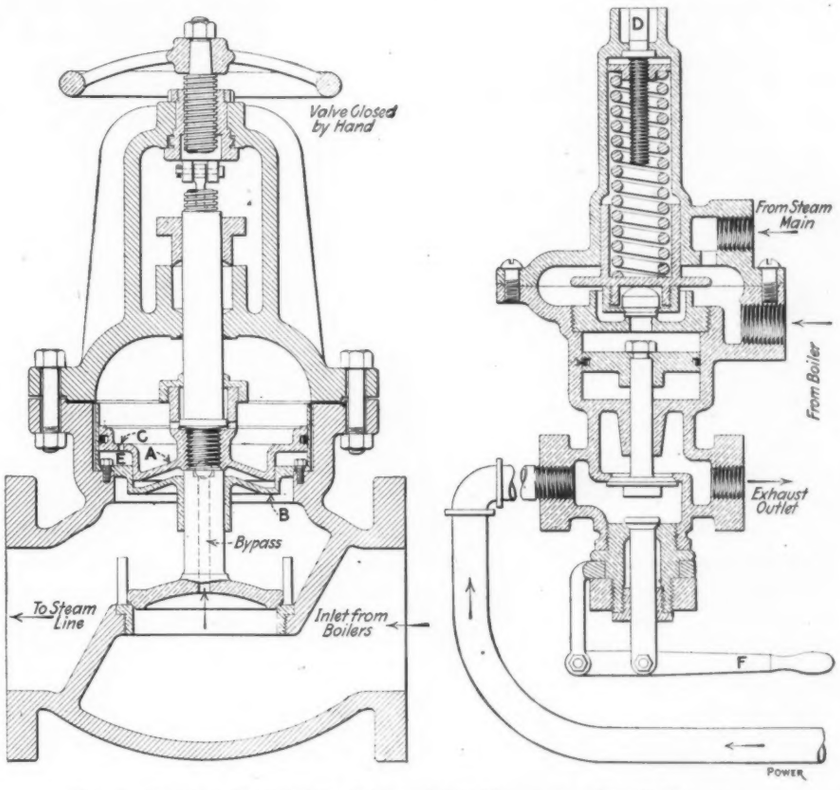


FIG. 1. DOUBLE CUSHIONED NON-RETURN VALVE AND PILOT VALVE

pipe connection leading to the pilot valve and if rupture occurs in the steam lines, pressure is removed from the top of the pilot valve, allowing it to open instantly, exhausting the pressure from between the dashpots *A* and *B*. The full boiler pressure always being above the dashpot, the valve will close at the same time being cushioned in the operation.

This valve can be opened or closed by hand, by raising the small lever *F*,

which releases it from the automatic position. The pilot valve, which is piped to the chamber *E*, can be set for any variation of pressure by adjusting the screw *D*.

Fig. 2 shows a double-cushioned, triple-acting, nonreturn globe valve with sealed bonnet. It is designed for the same purpose as the valve shown in Fig. 1. It also operates in connection with a pilot valve, similar to the sketch with an angle valve connected in the steam main.

A partial view of the pilot valve shows the diaphragm, adjustable spring and trigger pin. Full header pressure is carried by the pipe connection to the top of the pilot valve, where it compresses the spring in the central chamber, beneath the diaphragm, and so holds down the trigger pin extending downward outside to the pilot valve.

If a break occurs in the header the pressure above the diaphragm in the pilot is reduced, allowing the spring to lift the

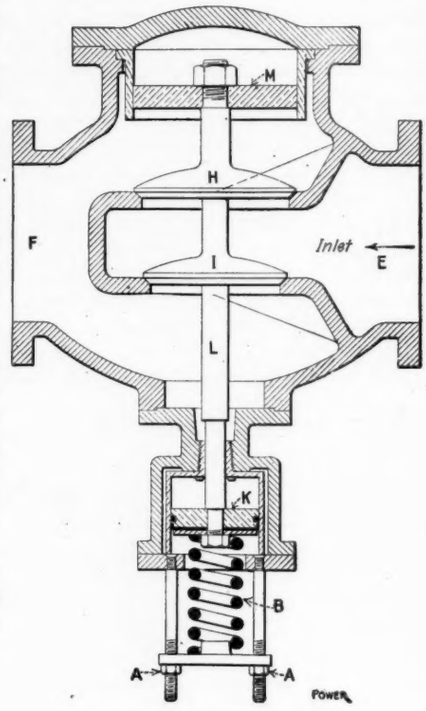


FIG. 3. REDUCING VALVE

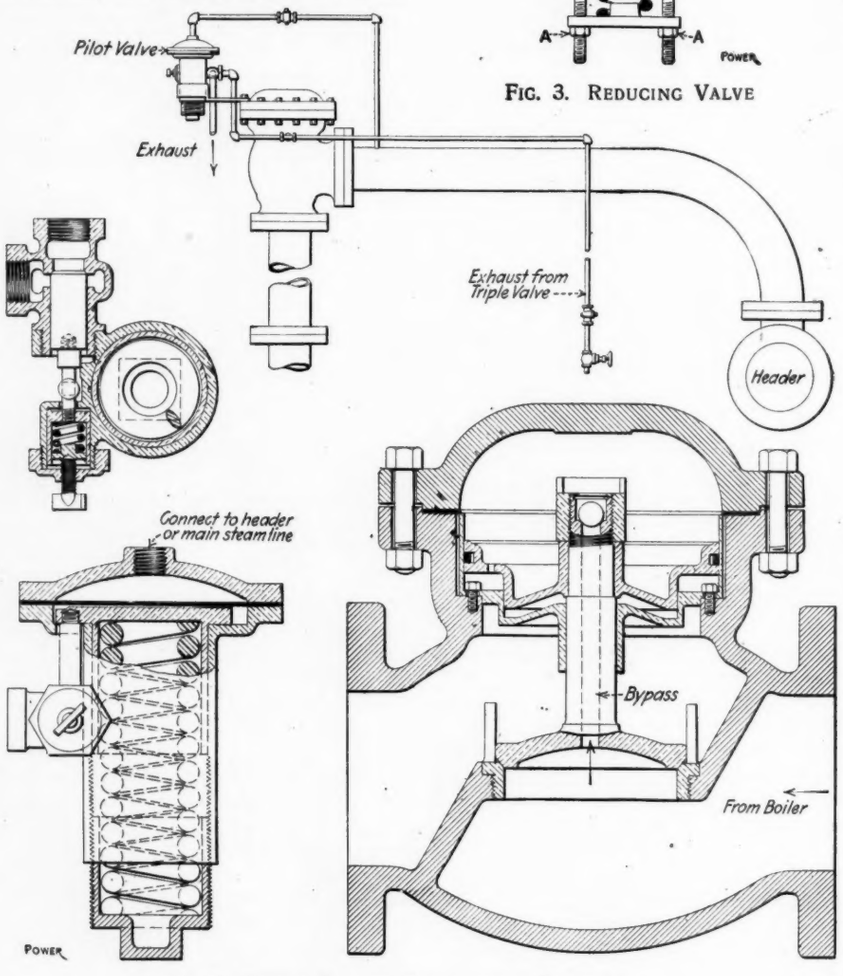


FIG. 2. TRIPLE ACTING NON-RETURN GLOBE VALVE

trigger pin and release the pilot valve. The boiler pressure from the cushion chamber of each triple-acting valve then has free vent to the exhaust opening in the pilot.

An automatic, double-seated, reducing valve for high and low pressure is shown in Fig. 3, in a closed position.

When the spring *B* is adjusted by the nuts *A* to the required pressure, the valve disks *H* and *I* are forced open. The high pressure or inlet is at *E*. The low pressure or outlet is at *F*. When the pressure on the low side has reached the pressure at which the spring *B* is set, the pressure still increasing, is exerted on top of the valve *H*, causing the valves *H* and *I* to close. The steam above and below the piston *M* and above the piston *K* prevents any chattering or hammering, and cushions the valve in opening and closing.

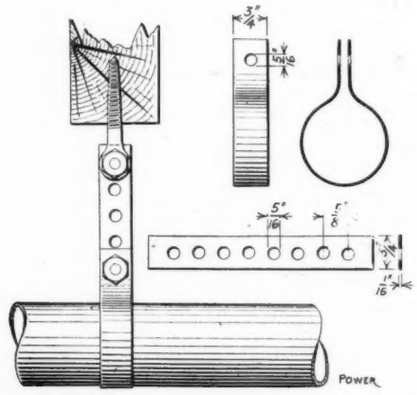
These valves are made by the Golden-Anderson Valve Specialty Co., Fulton Building, Pittsburgh, Penn.

Pipe Hanger for Light Piping

BY H. M. NICHOLS

The accompanying illustration shows the general details of a convenient pipe hanger for supporting light pipe, say from 1 in. up to 3 in. in diameter. It is made up of a V-shaped ¾-in. band-iron $\frac{1}{16}$ in. thick, and bent so that the inside diameter is the same as the outside diameter of the pipe it is to support. The flat ends of this piece are punched with $\frac{5}{16}$-in. holes.

The second part of the hanger is a flat strip of band-iron in which a number of $\frac{5}{16}$-in. holes have been punched about ⅝ in. apart. The strips can be



PIPE HANGER FOR LIGHT-WEIGHT PIPE LINES

made up in random lengths and when used broken off to the desired length.

Eye lag-screws are used to fasten the pipe hangers to the ceiling. These lag-screws are most easily put up before the pipe is put in place. The bands can be put on the pipe and clamped loosely to the straight strips with ¼-in. bolts. After the pipe is lifted into place it is only necessary to bolt the straight strips to the lag-screws with ¼-in. bolts.

This type of pipe hanger is particularly convenient when it is necessary to run a line of piping slanting.

Some tubes in one of the boilers of the French battleship "Verite" exploded recently, causing a panic among the sailors. Fortunately, nobody was injured.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Power Required for Pumping

How many horsepower must be developed to pump 200 gal. of water per minute against a pressure of 80 lb. per sq.in.?

D. P.

The weight of one gallon of water being 8⅓ lb., the weight of water pumped per minute would be

$$200 \times 8\tfrac{1}{3} = 1666.66 \text{ lb.}$$

A pressure of 1 lb. per sq.in. is equivalent to the pressure created by a column of water 2.309 ft. high, and pumping against 80 lb. pressure would be equivalent to pumping against a head of 2.309 × 80 = 184.72 ft. or raising the water to that height. The energy developed would be

$$1666.66 \times 184.72 = 307{,}865.43 \text{ ft.-lb. per min.}$$

or

$$\frac{307{,}865.43}{33{,}000} = 9.33 \text{ hp.}$$

Siphoning Feed Water Unreliable

What rate of flow can be relied upon for boiler feeding by piping water over a hill which is about 600 ft. distant and about 8 ft. lower than the source of supply, using 1¼-in. wrought-iron pipe, and is siphoning a reliable means of increasing flow?

E. W. H.

Using clean iron pipe 1¼ in. inside diameter laid without contractions or obstructions, a head of 8 ft. would overcome 600 ft. of pipe friction and deliver a flow of about 5½ gal. per min. at atmospheric pressure. A greater rate of flow could be temporarily secured by siphoning, but partial vacuum would be produced at the highest part of the siphon, causing air to become liberated from the water, and the pipe to become air-bound. Flow increased by siphoning should not be depended upon for direct boiler feeding.

Compression Space of Duplex Pump

What is compression space of the steam end of a duplex pump and how can it be measured?

G. D.

The steam ends of most duplex pumps have separate steam and exhaust passages, the latter being nearer the middle of the cylinder. When the piston passes over the exhaust port no more steam can escape, and whatever space then remains in the cylinder and steam passage will be compression space, in which any steam contained will be compressed and act as a cushion, opposing further motion of the piston until the end of the stroke. The volume of the compression space can be measured by moving the piston until the leading piston ring just covers the exhaust passage, and with the steam-chest cover and valve removed, finding the weight of water required to fill the compression space. The number of ounces of water so required divided by 0.5774 will be the volume of the compression space in cubic inches.

Pressure for Drying Kilns

What is the relative quantity of heat which can be transferred by coils of a drying kiln to be raised to 150 deg. supplied with steam at 100 lb. and when supplied at 130 lb. pressure?

J. J. G.

The temperature of dry saturated steam at 100 lb. boiler pressure (115 lb. abs.) is 338 deg. F. and at 130 lb. boiler pressure (145 lb. abs.) is 355.8, or practically 356 deg. F. The heat transmission of pipe coils with active circulation when supplied with steam at the pressures stated, would be at the rate of about 2 B.t.u. per hr. per sq.ft. of radiating surface for each degree difference between the temperature of the steam and the atmosphere to be heated.

Using steam at 100 lb. boiler pressure the transfer of heat per square foot of surface per hour therefore would be about (338 — 150) × 2 = 376 B.t.u. and on the same basis for steam at 130 lb. boiler pressure the transfer would be

$$(356 - 150) \times 2 = 412 \text{ B.t.u.}$$

or about 9½ per cent. more heat.

Weight and Thicknesses of Flywheel Rim

What should be the thickness of a rim for a cast-iron bandwheel 14 ft. in diameter with 24-in. face for a 20x42-in. Corliss engine running 90 r.p.m.?

E. R.

The formula for weight of flywheel is

$$W = 700{,}000\, d^2 s \div D^2 R^2$$

in which

W = Weight in pounds;

d = Diameter of cylinder = 20 in.;

s = Stroke = 42 in.;

D = Diameter of flywheel = 14 ft.;

R = Revolutions per minute = 90;

and substituting the values given, the weight would be

$$W = \frac{700{,}000 \times 20 \times 20 \times 42}{14 \times 14 \times 90 \times 90} = 7407.4 \text{ lb.}$$

As cast iron weighs 0.26 lb. per cu.in., the rim would contain 7407.4 ÷ 0.26 = 28,490 cu.in. As the face of the wheel is to be 24 in., if the section of the rim is rectangular and of uniform thickness, the area of one side of the rim would be 28,490 ÷ 24 = 1187 sq.in., and this latter quantity will be the difference in areas of two concentric circles, viz., one having its radius equal to the radius of the wheel, and the other having its radius equal to the inner radius of the rim. The radius of the wheel being 84 in., the area of the outer circle would be 22,167 sq.in.; the area of the inner circle would therefore be 22,167 — 1187 = 20,980 sq.in. and the inner radius of the rim would be

$$\sqrt{\frac{20{,}980}{3.1416}} = 81.72$$

or practically 81¾ in., and the thickness of rim would be

$$84 - 81\tfrac{3}{4} = 2\tfrac{1}{4} \text{ in.}$$

Combined Steam and Water Power

When the main jackshaft of a mill is driven by power developed by a waterwheel and an engine at the same time, what is the best way of throwing as much of the load on the water power as it will carry?

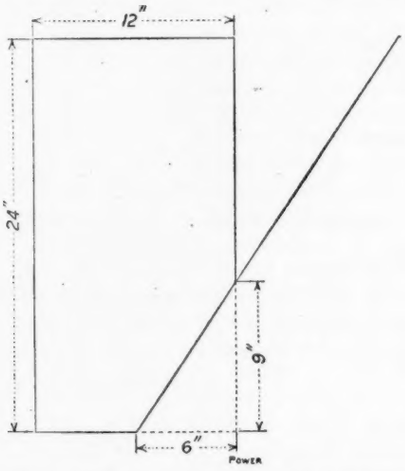
J. S.

To make the fullest use of the water power available, the engine governor should regulate the engine to a slower speed of jackshaft than the governor of the waterwheel, thereby calling on the engine only for power required in excess of the power developed by the waterwheel. In starting up, the load should be moved off by the power of the engine alone to better insure that the engine cylinder is warmed up and all water of condensation is removed before throwing on the power of the waterwheel. Neglect of the latter precaution has resulted in many serious engine smashes, especially during cold weather, when cylinder condensation is greatest.

Study Questions

This Week's Questions
Last Week's Answers

(116) What would be the volume of a cylindrical tank 12 in. in diameter and



24 in. high, made to fit against a sloping wall, as shown in the sketch?

(117) A steam-fitter went to purchase 1½ lb. of sheet packing and offered the correct change. The salesman cut a piece and weighed it; then remarked, "This runs 10c. over." The steam-fitter then told him to cut it in two, that he would take one-half of it and with the extra 5c. he would buy some putty. How much was the packing per pound?

(118) If 5½ lb. of water is heated from 50 to 75 deg. F., how many units of heat is absorbed and what is the equivalent in units of work?

(119) A pound of heated cast iron is plunged into 2 gal. of water and raises the temperature of the latter from 60 to 69 deg. F. What was the original temperature of the iron if its specific heat is 0.13?

(120) The thrust of a screw propeller is 10 tons, and acts against a ring having a mean radius of 6 in. The coefficient of friction being given as 0.06 and the shaft making three revolutions per second, how much power is consumed in friction?

Answers to the above will appear in the next issue. Answers to last week's questiions follow:

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(111) The increase in resistance due to an increase of temperature is 0.42 per cent. for each degree Centigrade above zero, the resistance at 0 deg. C. being taken as the base. Let

R_0 = Resistance at 0 deg. C.;
R_1 = Resistance at temperature t_1;
R_2 = Resistance at temperature t_2.

Then

$$R = R_0 (1 + 0.0042 t_1)$$

In the present case $R_1 = 100$ ohms; $t_1 = 90$ deg. F. $= 32.22$ deg. C.; and $t_2 = 75$ deg. F. $= 23.89$ deg. C. Substituting

$$100 = R_0 (1 + 0.0042 \times 32.22)$$
$$= 1.14\ R_0$$

whence

$$R_0 = \frac{100}{1.14} = 87.72$$

again

$$R_2 = R_0 (1 + 0.042\ t_2)$$
$$R_2 = 87.72\ (1 + 0.0042 \times 23.89)$$
$$= 96.52$$

or the resistance per foot at 75 deg. F. equals

$$\frac{96.52}{1000} = 0.09652\ ohms$$

(112) Let

x = Engineer's age when he took charge of the plant.

Then

$\frac{4x}{3}$ = Engineer's age now;

and

$\frac{x}{3}$ = Number of years since engineer took charge;

so that

$$\frac{x}{3} + 5 = \text{Oiler's age now} \qquad (1)$$

Let

y = Fireman's age when engineer took charge;

then

$$\frac{y}{2} + 2 = \text{Oiler's age now} \qquad (2)$$

and

$$y + \frac{x}{3} = \text{Fireman's age now} \qquad (3)$$

From (1) and (2)

$$\frac{x}{3} + 5 = \frac{y}{2} + 2$$
$$2x + 30 = 3y + 12$$
$$3y = 2x + 18$$
$$y = \frac{2x}{3} + 6$$

From this value of y in (3)

$$\frac{2x}{3} + 6 + \frac{x}{3} = x + 6 = \textit{Fireman's age now}$$

Let

z = Number of years before the oiler will be as old as the engineer was when he took charge.

Adding this to the oiler's age will equal the engineer's age or

$$\frac{x}{3} + 5 + z = x$$
$$x + 15 + 3z = 3x$$
$$3z = 2x - 15$$
$$z = \frac{2x}{3} - 5 \qquad (4)$$

In z years the sum of their ages is to be 150. Then

$$\left(\frac{4x}{3} + z\right) + \left(\frac{x}{3} + 5 + z\right) + (x + 6 + z) = 150$$

Substituting the value of z from (4)

$$\left(\frac{4x}{3} + \frac{2x}{3} - 5\right) + \left(\frac{x}{3} + 5 + \frac{2x}{3} - 5\right) + \left(x + 6 + \frac{2x}{3} - 5\right) = 150$$

Simplifying

$$\frac{14x}{3} = 154$$
$$14x = 462$$

$x = 33$ = Engineer's age when he took charge.

Then the oiler's present age is

$$\frac{x}{3} + 5 = \frac{33}{3} + 5 = 16\ yr.$$

and the fireman's age is

$$x + 6 = 33 + 6 = 39\ yr.$$

and the engineer's age now is

$$\frac{4x}{3} = \frac{132}{3} = 44\ yr.$$

(113) The total heat (above 32 deg. F.) in a pound of steam at 16 lb. abs. (from steam tables) is 1152 B.t.u. A pound of water at 115 deg. F. contains

$$115 - 32 = 83\ B.t.u.$$

A pound of steam condensed has therefore lost

$$1152 - 83 = 1069\ B.t.u.$$

Since

$$115 - 48 = 67\ B.t.u.$$

will raise 1 lb. of injection water from intake to discharge temperature it will require

$$\frac{1069}{67} = 15.955\ lb.$$

or nearly 16 lb. of injection water for each pound of steam condensed.

(114) The maximum unit stress developed upon the outer fiber of the shaft will be

$$s = \frac{16\ C\ d}{\pi\ (d^4 - d_1{}^4}$$

where

C = Twisting moment;
d = Outside diameter;
d_1 = Inside diameter;
$\pi = 3.1416$.

The twisting moment is the force P applied at a distance p from the center of the shaft or $C = Pp$. The problem gives $P = 3000$ lb., $p = 12$ in., hence

$$C = 3000 \times 12 = 36{,}000\ in.\text{-}lb.$$

and

$$d = 3\ in.\ \text{and}\ d_1 = 2\ in.$$

Substituting in the formula

$$s = \frac{16 \times 36{,}000 \times 3}{3.1416\ (3^4 - 2^4)} = \frac{1{,}728{,}000}{204.204}$$
$$= 8462.126\ lb.\ per\ sq.in.$$

(115) By the Law of Charles the pressure of a gas at constant volume varies as the absolute temperature

$350\ deg.\ F. + 461 = 811\ deg.\ abs.$
$32\ deg.\ F. + 461 = 493\ deg.\ abs.$

Then

$$811 : 60 :: 493 : x$$
$$x = \frac{60 \times 493}{811} = \frac{29{,}580}{811} = 36.47\ lb.$$

is the pressure of the air when the temperature is reduced to 32 deg. F.

Electric Railway Convention

The American Electric Railway Association and its affiliated associations held their thirty-first annual conventions in Chicago, Oct. 7, 8, 9, 10 and 11.

The report of the American Electric Railway Engineering Association committee on power generation possesses greatest interest for the majority of our readers. The first of the three subjects treated was "Method of Selecting Faulty High-tension Cables at the Power House." The replies to a letter sent to a number of companies operating large power plants, requesting information on their methods, indicated that some companies have not taken advantage of the development of the art and have no satisfactory method of selection. Others have methods fairly satisfactory to the local conditions. The replies in general showed a real need of further development of new ideas and apparatus to meet the requirements of satisfactory selection suited to their operating needs.

The second subject, "Chemical Laboratory Practice in Connection with Power Work," will be abstracted in a later issue.

The third, a paper by L. P. Crecelius, entitled "Boiler Settings," brought out that furnace arrangement has undergone considerable change recently and that it is now not uncommon to see a furnace projected out to a distance greater than the length of the stoker, to increase as much as possible the distance to the tubes from the point where the fuel is introduced.

In very recent installations the furnaces are installed entirely under the boiler instead of being extended, but the distance between the grate and the tubes is made very large to provide ample room for combustion.

In one case where horizontal water-tube boilers are employed, this distance is fully 12 ft.; in another, Stirling boilers provide an enormous combustion chamber. Therefore, in both cases the necessity of maintaining large and expensive arches is dispensed with. Almost every smoke-inspection bureau proposes, and in some cases insists upon, designs, including a horizontal baffle or arch, directly over the furnace, to correct improper existing conditions. It is believed, however, that if the boilers are set high enough in the first instance more freedom in the choice of arrangements for gas passes and baffling will be permitted.

In some recent tests upon archless furnaces operated continuously to produce 200 per cent. of the boiler rating, the tests being carried out for the purpose of determining limitations to successful operation at heavy overloads, Mr. Crecelius found that the ordinary firebrick linings of the furnace were subjected to enormous deterioration. Another difficulty developed in the fusing of ashes to the side and bridge-walls, just over the grate surface. A mass of fused ashes would accumulate until it extended all along the faces of the walls parallel with the grate, sticking out in some cases over 24 in., interfering with the forward movement of the fuel and proving extremely difficult to remove.

The rapidity with which such accumulations form depends upon the furnace temperature and the character of the fuel. A fuel having ash comparatively free of iron gives less trouble than one containing more iron; again, when boilers are operated at low rates and consequent low-furnace temperatures, the poorer fuel may be burned without serious trouble. However, because extremely high furnace temperatures are required, and in fact desired for smokeless combustion when boilers must respond to heavy overloads, all furnace linings are more or less subject to heavy deterioration.

The paper states the following conclusions: It is highly essential to provide large combustion chambers over furnaces intended for overloading, not alone for supplying space for flame, because with forced-draft furnaces this is unnecessary, but for the more important purpose of securing high initial temperature. A high temperature makes possible a minimum air supply to the furnace. Therefore, the cooling influence of the boiler surface or tubes must be sufficiently removed to prevent the rapid absorption of radiant heat, as only by this means can high furnace temperatures be secured.

Since the critical temperatures for ordinary furnace linings appear to have been reached already, it seems that future developments in the successful forcing of boilers must be accomplished by means of the archless type of furnace. This becomes more apparent when it is considered that vertical furnace linings are more easily and cheaply maintained than the large horizontal, overhanging surfaces or arches.

The following officers of the American Electric Railway Association were chosen on Oct. 10: President, George H. Harries; first vice-president, Charles N. Black; second vice-president, C. Loomis Allen; third vice-president, C. L. Henry; fourth vice-president, John A. Beeler.

On Oct. 11 the American Electric Railway Engineering Association elected as president, Martin Schreiber; first vice-president, L. B. Crecelius; second vice-president, John Lindall; third vice-president, J. H. Hanna; secretary-treasurer, H. C. Donecker.

The large German cruiser "Goeben," which was launched last year, is now undergoing her trials. It is reported that she has attained a speed of 32 knots.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Already, we are ready to place R. Reddie, of Newmarket, Tenn., in a choice niche of our Hall of Fame. Who is next?

Extensive improvements are shortly to be made at the paper mill, prominent among which will be the installation of a 700,000-hp. boiler—Steubenville (Ohio) "Herald-Star."

Our friend Frazier sends us this clipping, and adds: "Where would Steubenville be if the thing exploded?" We know no one able to answer this question, unless it be Danny Hogan or possibly the angel Gabriel.

Whatever may be said of our water-power possibilities, our power on water is a grim reality. The fleet recently in New York waters is capable of developing 427,000 hp. Uncle Sam's only near rival seems to be that 700,000-hp. boiler over at Steubenville.

Don't be misled by that wily refrigeration engineer who claims that half an hour spent in a brewery refrigerating room will cure a stubborn case of hay fever. As your Uncle Spill diagnoses the case, there's small choice between a sneeze and a hiccough—and he's tried 'em both.

A Hamburger has staked his reputation on a tide-power scheme to provide the German consumer with two-cent current by utilizing the North Sea as a power plant. This sprightly little H_2O body ought to make a fair ice-olated plant in winter time.

The House of Hill is to have a home of its own. The family has so greatly outgrown the old homestead on the Pearl St. farm that our Old Man, seeing his chance on Tenth Ave., just dug down into his old woollen stocking and closed the dicker. Soon we will start clearing away the brush for a home that will make our neighbors stand bolt upright and throw down their hoes. We will have all the fixings that this village's skill can provide. Along about February, 1914, when the sledding is good and the last shingle is nailed on the roof, fish out your woollen mittens, wrap yourself and the missis in the buffalo robe, and start old Dobbin jingling the bells right up to our dooryard. This is you-all's "invite" to the housewarming. If interested, read about it on page 626.

Minneapolis N. A. S. E. "Booster" Meeting

Minnesota Association No. 2 of the National Association of Stationary Engineers held a "booster" meeting and banquet at the Hotel Radison, Minneapolis, on Saturday evening, Sept. 28. It was the largest and most enthusiastic gathering in the association's history.

Minnesota No. 7, of St. Paul, came over in a body, and employers and many leading citizens helped to make the attendance nearly 300. The banquet was in the hotel ball room.

Prof. J. J. Flather, of the University of Minnesota, delivered an instructive and witty discourse on engineering matters in general.

National President J. F. McGrath, of Chicago, told of the aims and objects of the association and what it has accomplished in the past and hopes to do in the future. He assured No. 2 of his hearty support at all times during his administration.

J. A. A. Burnquist, a member of Minnesota No. 7, a candidate for lieutenant-governor of the state, and one of the association's best friends, discussed license laws, convincing all present that he knew his subject and is working along the proper lines.

John W. Lane, Chicago, editor of the *National Engineer*, satisfied his hearers that he is the right man to manage the national organ, and his advice and counsel made a deep impression.

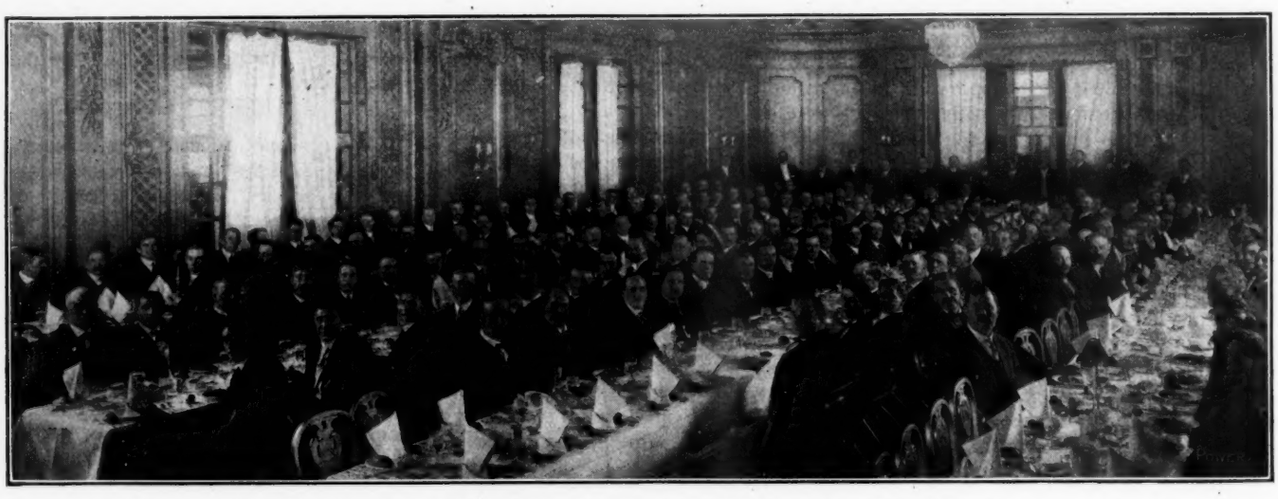
W. L. Harris, president of the New England Furniture & Carpet Co., a director of the Civil and Commerce Association, and one of Minneapolis' most successful citizens, gave a characteristically vigorous and spicy talk. The toastmaster then called for applicants for membership and 40 could not resist the temptation.

Alderman C. D. Gould, a candidate for mayor, spoke of the great good accomplished by the association and offered his services whenever he could be of assistance.

Other speakers were J. B. Crane, Duluth, state president; Herman Mueller, Minneapolis, state vice-president; James McCreary, St. Paul, state secretary; T. S. F. Hayes, St. Paul, state deputy and president of No. 2; Fred Camitsch, St. Paul, and J. M. Williams, Minneapolis.

The meeting and banquet, conducted by the entertainment committee, J. F. Gould and Frank Rosenthal, was such a complete success that it is planned to make it an annual event. This is the beginning of an active campaign for new members and No. 2 expects to have every engineer in the city enrolled in the near future. The educational and entertainment committees are already planning some very interesting things for this winter.

In the afternoon the national officers were driven about the Twin Cities in automobiles accompanied by members of No. 2 and shown some of the large power plants and the many points of interest.



BOOSTERS AND BANQUETERS AT THE MINNEAPOLIS N. A. S. E. MEETING

New York Electrical Show

The usual annual electrical exposition and automobile show was held at the Grand Central Palace, New York City, from Oct. 9 to 19. There were about 100 exhibits on the three floors of the building and the large daily attendance showed that interest in this kind of exposition has not lagged.

A luncheon to Thomas A. Edison, commemorating the completion of 30 years of central-station service, opened the exposition. J. W. Lieb, Jr., acted as toastmaster and Samuel Insull, of the Chicago Commonwealth Edison Co., was among the speakers.

The central stations in and around New York City were well represented, and the U. S. Government contributed several instructive exhibits, that of the electric class at the Brooklyn Navy Yard being especially interesting.

Institute of Operating Engineers

At the first meeting of the new council of the Institute of Operating Engineers, L. Houmiller was appointed executive secretary in place of H. E. Collins, whose term had expired and who is too busy to give the necessary time this year. Mr. Houmiller has been with the institute since its inception and is ready to do all he can to advance the work and help accomplish the aim of the institute.

The constitution, as revised at the annual meeting in September, will be printed and circulated the latter part of this month. The *Bulletin* will be out the early part of November. A copy will be sent to those interested by writing to the secretary, 29 West Thirty-ninth St., New York City.

Our New Home

The Hill Publishing Co., John A. Hill, president, will erect a 14- to 16-story building at the northeast corner of Tenth Ave. and Thirty-sixth St., New York City. The property comprises half the block on Tenth Ave., and extends 175 ft. in Thirty-sixth St., and is covered with tenements.

It is Mr. Hill's object to have the most complete publishing and commercial building in the city. Every known convenience for cleanliness, comfort, safety, low insurance and economical conduct of business will be installed. The publishing company will occupy several floors. A special feature of the building will be a roof garden for the recreation of the occupants of the building, both tenants and employees. It is expected the building will be ready to occupy February, 1914.

Boiler Accident in Poughkeepsie Plant

At 10 p.m., Sunday, Nov. 13, a tube pulled out of the mud drum of a Stirling boiler operating under 160 lb. pressure at the plant of the Central Hudson Gas & Electric Co., Poughkeepsie, N. Y. Four men were badly injured, but Fireman Doyle, who was reported dying, is rapidly recovering. Due to a loss of steam pressure caused by the ruptured tube, a part of the service was interrupted until another boiler was cut in on the line. The interruption was trivial, however, as the load carried at the time of the accident was rather light.

The boilers are equipped with Murphy furnaces, and Fireman Doyle had the firedoor open breaking a clinker when the tube let go. Another man was in the front ashpit of the boiler directly opposite and both were deluged with incandescent fuel and steam and water. An attendant who rushed to the top of the boiler to shut the stop-valve was severely burned about the hands, and the engineer, in an effort to rescue the injured, lost his way in the steam-filled room and fell into an open outside ashpit, receiving bruises about the limbs and back.

The tube that let go was almost directly in front of the fire-door, and in pulling out of the mud drum it assumed an angle directly opposite to that at which it entered the drum, or, instead of the open end facing the drum it pointed away from it, straight toward the fire-door. The doors of a Murphy furnace are very heavy, and if they had been closed when the tube pulled out, it is possible no one would have been injured as the steam pressure might have dissipated itself in the furnace, the volume of which is very great, without blowing the doors open immediately, thus giving the attendants time to get out of the path of the hot fuel and steam and water.

It is not definitely known whether the cause of the accident was corrosion of the tube or faulty workmanship in securing it to the drum.

Hot Water Tank Explosion

Shortly after 3 a.m., Oct. 2, a hot-water tank exploded in a café in New Bedford, Mass. The interior of the building was badly wrecked and the plate-glass front blown across the street. The damage amounted to about $5000.

Ten-inch floor timbers were split from end to end, a mezzanine floor laid on 3 in. of reinforced concrete was torn up, marble-topped tables were thrown to the ceiling and smashed. While all this occurred over the tank and heater, bottles and glasses remained undisturbed on the bar 15 ft. away.

The tank which exploded was of 75 gal. capacity, of heavy copper with soldered and riveted seams and marked with a 500-lb. test mark. It furnished hot water for the kitchen and was suspended from the basement ceiling. Below the tank was a cast-iron heater which was uninjured. The tank was connected to the city mains, the pressure in which automatically kept the tank full.

The tank had a lever-safety valve which an employee stated had never been in working order. After the explosion, the cold-water supply pipe was found to contain a check valve which prevented relief of the excess pressure from expansion into the street mains when the safety valve failed to operate.

NEW PUBLICATION

PRACTICAL AERONAUTICS. By Charles B. Hayward; with introduction by Orville Wright. American School of Correspondence, Chicago, Ill. Cloth; 6½x9¾ in.; 769 pages; 310 text figures. Price, $3.50, net.

While this work is necessarily somewhat bulky, it is a well digested and readable compilation on aëronautics. Sufficient theory is simply treated to aid the reader in readily understanding the various types of construction. Very little space has been given to dirigible balloons, which are for the most part confined to the Wellman, Vaniman and Zeppelin ships.

Under "Types of Aëroplanes" are described the "standard" biplanes of Wright, Curtiss, Voisin, Farman and Cody; the "standard-type" monoplanes include the Antoinette, Santos Dumont, Bleriot, Grade and Pfitzner types.

The section devoted to building and flying an aëroplane will be of undoubted interest to many readers. It includes the construction of models, a glider, Curtiss biplane, Bleriot monoplane, the art of flying, accidents and their lessons, and amateur aviators.

While it will be some time before aëronautics becomes a popular pastime of the operating engineer, to those who are interested, and can afford to fly, this book contains much valuable information.

(Locomotive)—In the notice of the bound volumes of *Locomotive* for 1910-1911, which was printed in the Oct. 8 issue, the mistake was made of saying that these volumes are complimentary. The price of the bound volumes is $1 each.

OBITUARY

On Sept. 18, Frederic Austin Warren, for 11 years chief electrician of the Colorado Fuel & Iron Co., died at Canon City, Colo. Prior to his connection with this company, he was employed by the General Electric Co., and later served as electrician at the Fremont and Coal Creek mines.

PERSONAL

J. O. Westberg has assumed charge of the mechanical department of the Felber Engineering Works, New York City. Mr. Westberg was formerly with the Sibley Construction & Supply Co., New York City.

Arthur Hadley, managing director, and Bernard Price, chief engineer, of the Victoria Falls & Transvaal Power Co., South Africa, recently arrived in America, and will visit some of the large electric-power plants.

Truman D. Hayes has resigned as New England sales engineer of the Triumph Electric Co., Cincinnati, Ohio, to accept a similar position with the International Steam Pump Co., New York City., which he will represent in eastern New York.

Kern Dodge, a well and favorably known Philadelphia engineer, has returned from a year's travel abroad and will open an office in the Morris Building, Philadelphia. He will devote himself henceforth to the engineering and financing of public-service properties. Mr. Dodge was a founder and for many years a partner of Dodge & Day, engineers.

Frank Koester, consulting engineer, has removed his office from 115 Broadway to larger quarters at 50 Church St. (Hudson Terminal Bldg.), New York City. Mr. Koester is principally engaged in steam and hydro-electric power-plant work, electric transmission and traction. He was recently retained as an expert by the Borough of Manhattan, New York City.

Wentworth P. Johnson has been appointed New York manager for H. M. Byllesby & Co. For the last nine years he was the European financial agent of the International Harvester Co., residing in London. Mr. Johnson is an old friend and business associate of Mr. Byllesby, having been connected with him in the Northwest General Electric Co. at St. Paul.

Carlisle Mason, vice-president and general manager of the Nelson Valve Co., Philadelphia, Penn., has tendered his resignation, to take effect Nov. 1, 1912. He assumed charge of this company five years ago and built up the business to its present standing, establishing the model plant at Chestnut Hill, Penn. Mr. Mason is well known in the trade. On his return from a trip of several months abroad, he will probably reënter the valve and fitting business.

The Ridgway Dynamo & Engine Co. is building the Rateau turbine, under arrangement with the American representatives of the inventor. A 1000-kv.-a. unit built at the shop has been in successful operation for several months and a number of other units are now in hand.

Moments with the Ad. Editor

You've been hearing a lot lately about "conservation of energy."

The difference between a great and an insignificant man lies in the amount of *reserve power* he has back of him.

* * *

Napoleon, probably the world's greatest modern soldier, never let his "old guard" go into a fight unless the exigency was most urgent—he held them in *reserve* and then, at the critical moment, they swept all before them. His plan succeeded admirably until he met a man with *more reserve force* than he had.

The lungs contain a *reserve* of air that is never used—the lymphatic system is nature's reserve of *all* the forces and elements used in the human body. Men have gone without food for long periods of time simply existing on the *reserve* stored up in that lymphatic system.

The government *reserves* great forests and waterways for use by a future generation—wise men put money in *reserve* against a "rainy day." It is the *energy* we have *conserved*, lying dormant, that allows us to meet a crisis and *overcome* it.

Now consider conservation of thought and *you*. Sometime in your life an opportunity has come to you. If you've grasped it, it was only because you had the "know how"—the *reserve knowledge* to bring to bear. Another opportunity will arise probably at no far distant date, have you a reserve to meet *that*?

What is the answer? Study. And what you learn, *conserve*. Books, lectures, periodicals, all are excellent, but in *your* case, you men whose future positions depend on the knowledge you have of your work and its equipment—you should study *advertisements*.

In them you receive first news of the newer, the greater improvements in cost-cutting, result-pulling appliances. First hand information of the fight that's continually going on to improve *conditions*. And you can *depend* on the news you get in an advertisement absolutely for, through experience, manufacturers have learned that it is ruinous to them to advertise dishonestly.

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So study advertising, think how it applies to *your* case—consider the products—find out if they'll *help* you—and *remember* what you read.

Some day, when the *need* arises, you'll have *conserved* knowledge—enough to *meet* the need.